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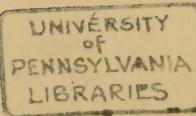
AN INVESTIGATION AND DISCUSSION OF CYLINDER CONDENSATION.

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Harold T. Moore.

June 1901.

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H. F. S.:
S. G. Lewis
14 May 1904
20 May

The object of this thesis is an investigation of the heat changes in a steam engine, with an attempt to form some idea of what causes the phenomenon known as initial cylinder condensation and how it takes place. We know that the losses in an engine, due to this cause are very large and excessive. Possibly if we knew just what happened in the steam engine cylinder itself to cause this initial condensation, some means might be devised to prevent or reduce it and thus make a very great saving in the cost of steam power, the engine being by this means made

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to give much more economical results.

We propose to give the thesis under the following heads.

- 1- Outline of what was done, and the methods of taking the necessary readings. p. 4
- 2- Instruments used with their descriptions, constants, and calibrations. p. 8
- 3- Engine used in making the tests, with its constants. p. 34
- 4- Steam tables etc. used in working up results. p. 37
- 5- Calculations of test results; I.H.P., B.H.P., eff., etc. p. 38
- 6- Diagrams, with their explanations and the methods used in making them. p. 46
- 7- Method employed to draw

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curves showing the variation of heat with time. p. 62

8- Curves between heat and time, with their explanations, etc. p. 67

9- Sources of error in these calculations and curves. p. 80

10- Possible theories and conclusions. p. 83

1- Tests were made on a steam engine, with readings taken to determine the following quantities: Steam consumption; revolutions per minute; quality and pressure of steam supplied; quality and pressure of the steam in the

exhaust. I incidentally took these readings were taken to determine the I.H.P., B.H.P., and quantity of heat given up in the condenser. Indicator cards were taken from each end of the cylinder every fifteen minutes.

The method employed to determine these results was as follows: the engine was run with a load applied to the fly-wheel in "by means of a Prony brake, this load being kept constant and measured by the pressure of the brake upon a set of weighing scales; indicator cards were taken by means of a pair of "Thompson" indicators

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connected to the ends of the engine cylinder; the revolutions of the engine were determined by means of a recording revolution counter, worked by a string attached to a pin in the end of the shaft; the quality and pressure of the steam in the supply and exhaust pipes were determined by means of two steam pressure gauges and calorimeters which took steam from the respective pipes; in order to determine the steam consumption of the engine, the exhaust steam was condensed in a surface condenser at atmospheric pressure and then weighed; the

quantity of heat given up in the condenser was determined by measuring the temperature of the condensed steam, the amount of condensing water, and its rise in temperature while passing through the condenser.

From the data thus obtained, curves were drawn to show the following: quality of the steam in the cylinder at each point of the stroke during expansion; variation with time of the quantity of heat held by the steam in each end of the cylinder; variation with time of the amount of work done and also of the quantity of heat radiated to

the outside air; variation with time of the amount of heat supplied to the engine, and also the amount rejected by the engine. A study of these curves shows what probably happens in the cylinder to cause this initial condensation effect.

2- The brake used to apply the load was of the ordinary Prong type with four turns of rope around the fly-wheel, the frame being made of iron pipe. The zero reading of the weight of the brake on the scales was determined in the following

manner. The end of the brake being rested on the weighing scales, the engine is first turned, by hand, slowly forward and the reading of the pressure on the scales taken. The engine is then turned backward in the same manner and the reading of the scales again taken.

The mean of these two readings is the true scale reading when the engine is standing at rest. This method of getting the zero point of the scales, eliminates the effect of friction; since the friction force acts in the second case opposite to what it does in the first. Brake arm is 106 inches.

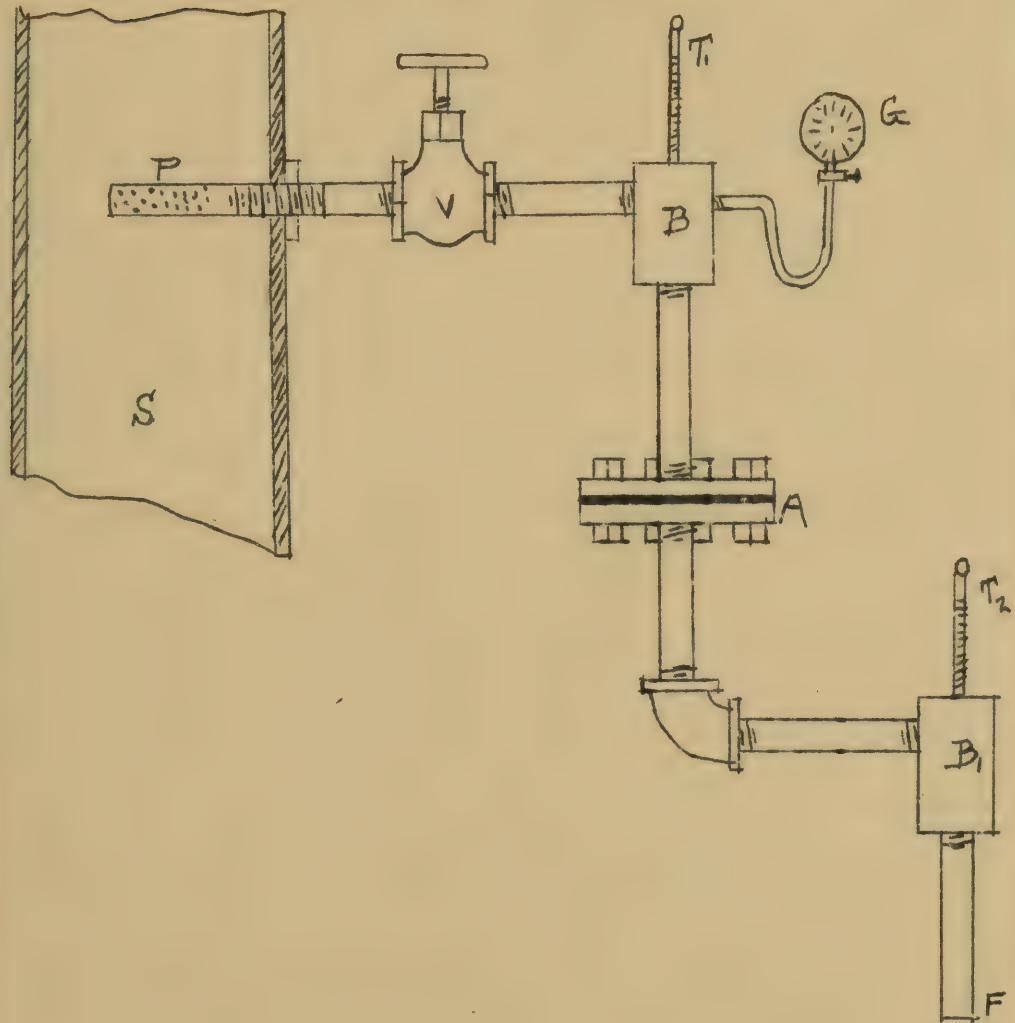
The revolution counter was

of the recording type, made by "Schaeffer and Budenberg" of New York. It was fastened to a wooden block that rested on the floor of the engine room. The counter was worked by means of a string which hooked over a pin in the end of the engine shaft. Readings of the counter were taken at stated intervals of time.

A "Barus Universal" steam calorimeter was used to get the quality of the steam supplied to the engine. This instrument is described in the "Transactions of A. S. M. E." vol. XI, p 790. As the moisture in the steam used in this test was small only the

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heat gauge part of the calorimeter
was used. A sketch and
description of this is given
below:



S is steam pipe which supplies engine.
P is a sampling pipe which serves to take steam from pipe S. It is closed at the end, has many small holes in its sides, and extends into the centre of the pipe.
B and B₁ are thermometer cups into which set two thermometers, T₁ and T₂.

V is a valve to control the steam supplied to the calorimeter.

G is a steam gauge which gives the pressure of the steam supplied. Let this pressure be equal to p_1 .

Steam to the calorimeter passes through an orifice at A and out into the atmosphere at F. Its pressure is reduced while passing through the orifice from that at G to the atmospheric.

The two parts of the calorimeter are insulated so that no heat is conducted by the pipe from the upper to the lower part of the instrument.

If, x = dryness of steam.

q = heat of liquid.

r = " " vaporization.

Then the heat in one pound of steam, above the orifice A, is:

$$q_p + x r_p = H.$$

After passing the orifice, the pressure being lowered, the steam is superheated by its own heat. Hence the heat in one pound of steam, below the orifice is:

$$q_a + r_a + .48(t_1 - t_2) = H_1.$$

Where a = atmospheric pressure.

$t_1 - t_2$ = degree of superheat.

Equating these two heats, since it is assumed that no heat is lost in passing through the orifice we have:

$$q_{p_1} + \lambda n_{p_1} = q_{p_2} + n_a + .45(t_1 - t_2) \quad (I)$$

t_2 = saturation temperature = 212°F .

t = temperature of steam at B.

The temperature read by the thermometer T_2 is however lower than t_1 , since the thermometer is not in direct contact with the steam. Similarly the reading of thermometer T_1 is lower than the temperature T of steam at the pressure p_1 .

It is assumed that

$$t_1 - T_2 = T - T_1$$

T before

$$t_1 = T - T_1 + T_2.$$

Hence from equation (I).

$$x = \frac{q_a + r_a + 48(T - T_1 + T_2 - t_2) - q_p}{r_p}.$$

This gives the quantity of the steam in the supply pipe.

The gauge, \mathfrak{G} , which was used to get the steam pressure p , is "B andor" steam gauge no. 1874 56, made by "Williamson and Cassidy" of Philadelphia. Its scale is graduated in pounds from 0 up to 120 pounds, gauge.

This gauge was calibrated by means of a "Navy" steam gauge-tester. From the calibration table the curve was drawn.

Calibration Curve
of
Gauge No. 189436.

Scale inv. = 5/lbs.

True Readings.

105

100

95

90

85

80

90

100

105

110

115

Gauge Readings (mean).

This gauge was calibrated for both ascending and descending readings, the mean being given on the curve as the "gauge-reading". The gauge pressures were corrected from this curve and the true pressures used in working up results.

A "Carpentier" separating calorimeter was used to get the value of x in the exhaust. It took steam from the exhaust pipe by means of a sampling tube similar to the one used with the Bunsen calorimeter. This instrument is completely described in "Carpentier's Experimental Engineering" in 402, edition of 1897; also in the catalogue of "Schaeffer and Budenberg",

pp. 105, edition of 1899. The water separated out is measured by means of a graduated scale placed alongside of a gauge glass connected with the inside of the instrument and in which the height of the collected water shows. Each scale division represents .01 pound of water. The amount of steam going through the instrument is measured by being passed through an orifice. The weight of steam flowing through is determined from the formula of Nafin, which is:

$$W = \frac{\rho \cdot A}{70}$$

Where; ρ = absolute press. of steam
 A = area of orifice.

When steam flows through an orifice of area A from a pressure P_1 to a lower pressure P_2 , the weight of steam flowing through in a given time is $W = P_1 \times K$, K being a variable.

The formula of Napier gives:
 $K = \frac{A}{70}$, when W = pounds of steam through the orifice per second, and P_2 is less than $\frac{3}{5} P_1$.

As this calorimeter was used where P_2 was atmospheric pressure and P_1 was just a pound or two higher, it was necessary to calibrate this calorimeter to find how the value, K , varied with the pressure P_1 .

This was done by comparing the two gauge scales mentioned and described on page 19. This method gave the value of K for

values of P , above 25 pounds, absolute. For lower values of P , than this the following method of calibration was employed: the pressure P , was measured by means of a mercury manometer; the steam discharged through the orifice, A , was condensed and weighed. Thus by taking the time of discharge it is very simple to find the mean rated discharge through the orifice A for various low pressures, P .

The curve on page 18-C shows the variation of K with the pressure P .

In using the values from this curve, W = pounds of steam discharged in 10 minutes.



The gauge attached to this calorimeter had two scales, one reading pounds pressure, and the other reading pounds of steam through the orifice in ten minutes.

From a comparison of the two scales, the constant $\frac{A}{70}$, was found to be .04; where W is the number of pounds of steam through the orifice in ten minutes. The gauge was graduated down to about 20 pounds absolute. As the pressure in the exhaust was only about 15 pounds absolute, a mercury manometer, graduated in inches, was used to measure the steam pressure at the calorimeter. At the lowest graduation on the

gauge scale, the constant was found to have dropped only to .038 instead of .04.

Because of this fact, and since it was found that a very considerable error in the amount of steam through the orifice made but slight error in the value of x , it was assumed that the constant, $\frac{A}{70}$, was always equal to .04, even at the low pressure of 15 pounds absolute.

If, n = pressure in inches of mercury,

then $\frac{n}{2}$ = pressure in pounds,

.47 = atmospheric pressure.

$\therefore [14.7 + \frac{n}{2}] .04 = W$, the pounds

of steam through the orifice

in ten minutes.

m = weight, in pounds, of the water separated out by the calorimeter in ten minutes. It is determined from readings of the height of the water in the gauge jars.

Then we have:

$$x = \frac{W - m}{W + m} = \text{dryness of straw.}$$

The condensed straw from the condenser was caught in a tank placed upon a set of weighing scales. By reading these scales at stated intervals of time, the weight of straw taken by the engine per minute is easily determined.

The condensing water was measured by being passed over

a standard weir having end contractions. The head of water on the weir was measured by means of a hook gauge placed in the weir flume. The weir head was 10 inches wide. Zero reading of the hook gauge was taken when the water level in the flume was just at the crest of the weir.

Formulae used in connection with the weir were those of "Francis":

$$Q = 3.33 (l - .2h) h^{\frac{3}{2}}$$

Q = cu. ft. of water over weir per second.

l = width of weir, in feet.

h = head of water on weir, in feet.

Thermometer cups were placed in the pipes just

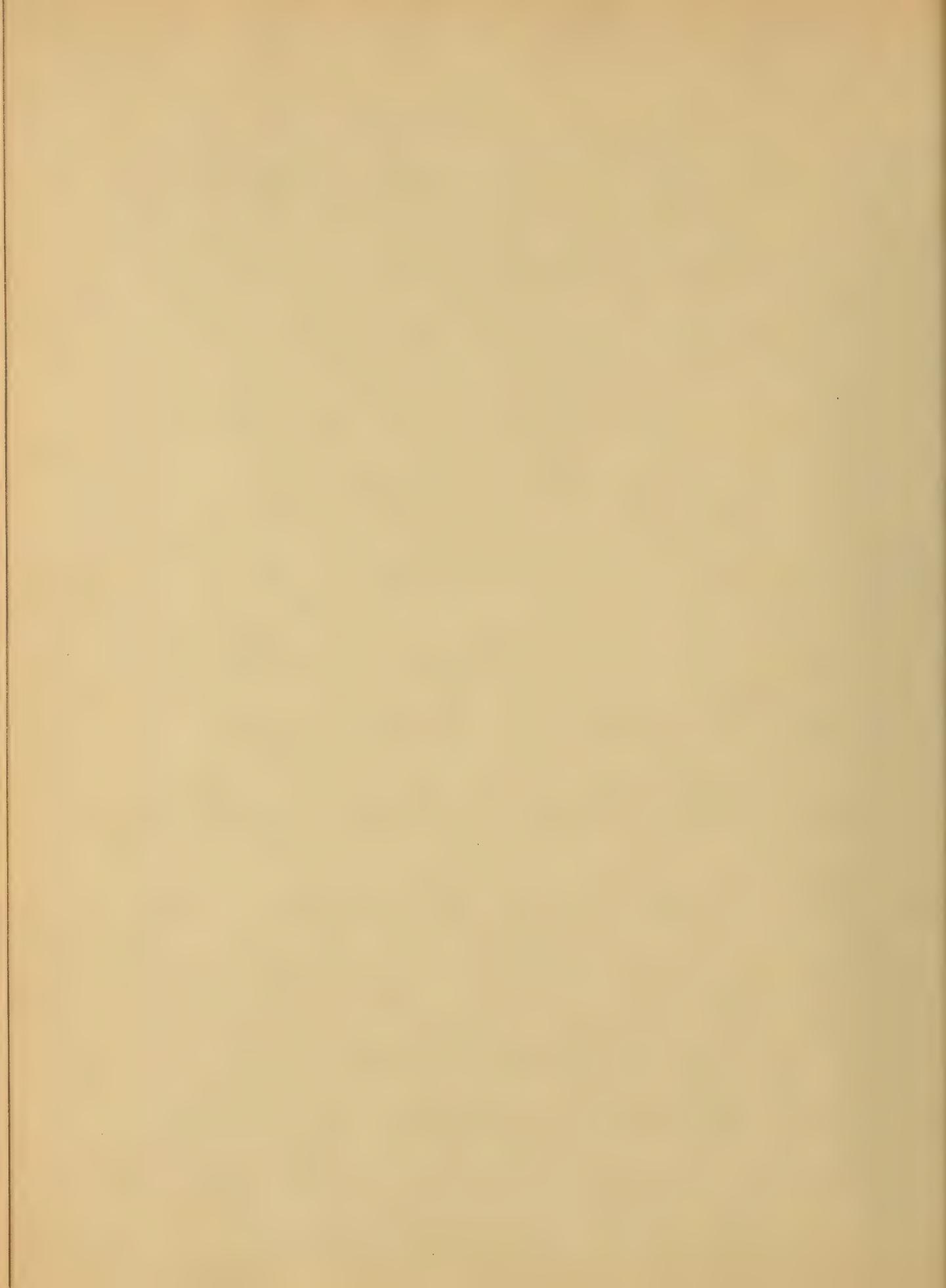
outside of the condenser. By means of thermometers placed in these, the following temperatures were taken; condensed steam, condensing water entering condenser, condensing water leaving condenser.

These thermometers were graduated to read Fahrenheit degrees. From these temperatures, knowing the amount of condensed steam from the condenser and also the amount of condensing water, it is a very simple matter to find the amount of heat given up or accounted for in the condenser.

The indicators used to take cards with over two "Thompson Improved" instruments,

made by the "American Steam Gauge Co." of Boston. This form of indicator is familiar to every engineer, so a description of it is considered unnecessary. However, a very good description is given in the catalogue of "Schaeffer and Budenberg", page 78, edition of 1899. Only a few details are different in this description from those on the actual instruments used in making this test. These indicators were fastened on to cocks set directly in the sides of the engine cylinder, with no pipe connections whatever.

The springs used in these indicators were 60 pound "Thompson"



springs, numbered the same as the indicators themselves.

Indicator & spring no. 3257 was used on the head end of the engine cylinder, and no. 3071 on the crank end.

These springs were calibrated in their respective indicators by a method of comparison with a strain pressure gauge. This method is described in "Carpenter's Experimental Engineering", page 479, edition of 1897. It consists in applying the same strain pressure to the indicator cylinder and to the strain gauge, thus comparing the gauge reading with the amount of movement of the indicator pencil.

The strain pressure was

made to gradually rise, starting at the atmospheric after the instruments had become heated up. For every increase of 10 pounds in the strain pressure as read by the gauge, the indicator drum was turned, thus making the pencil draw a horizontal line. This gave the "ascending" calibration.

A similar process was employed with the pressure gradually descending again to the atmospheric. These horizontal lines show the amount of vertical movement of the indicator pencil corresponding to the different strain pressures as read by the gauge.

These horizontal line calibrations are given on pages 29 and 30, the strain pressures marked being those read by the gauge used, which was of the "Boudon" type, no. 159434, made by "Williamson and Cassidy" of Philadelphia.

This gauge was then tested as a "Crosby strain gauge Tester" and its ascending and descending calibration curves drawn as on page 31. By measuring the amount of strain compression (actually, final movement) from the diagrams on pages 29 and 30, and then correcting the strain pressures by means of the curve on page 31, we are enabled to

draw both "ascending" and "descending" curves showing the variation of spring compression (pencil movement) with the true strain pressure under the indicator piston. These curves are then used to convert the indicator cards taken by a method to be described later.

These calibration curves are given on pages 32 & 33 of this thesis.

Calibration of Indicator Spring No. 3257

Pres. (lbs)	Compression (In.)
95	1.44
90	1.35
80	1.18
70	1.02
60	.84
50	.66
40	.49
30	.35
20	.22
Atm.	0

Ascending Readings.

Pres.	Comp.
95	1.50
90	1.42
80	1.27
70	1.10
60	.93
50	.75
40	.59
30	.41
20	.24
10	.06
Atm.	0

Descending Readings.

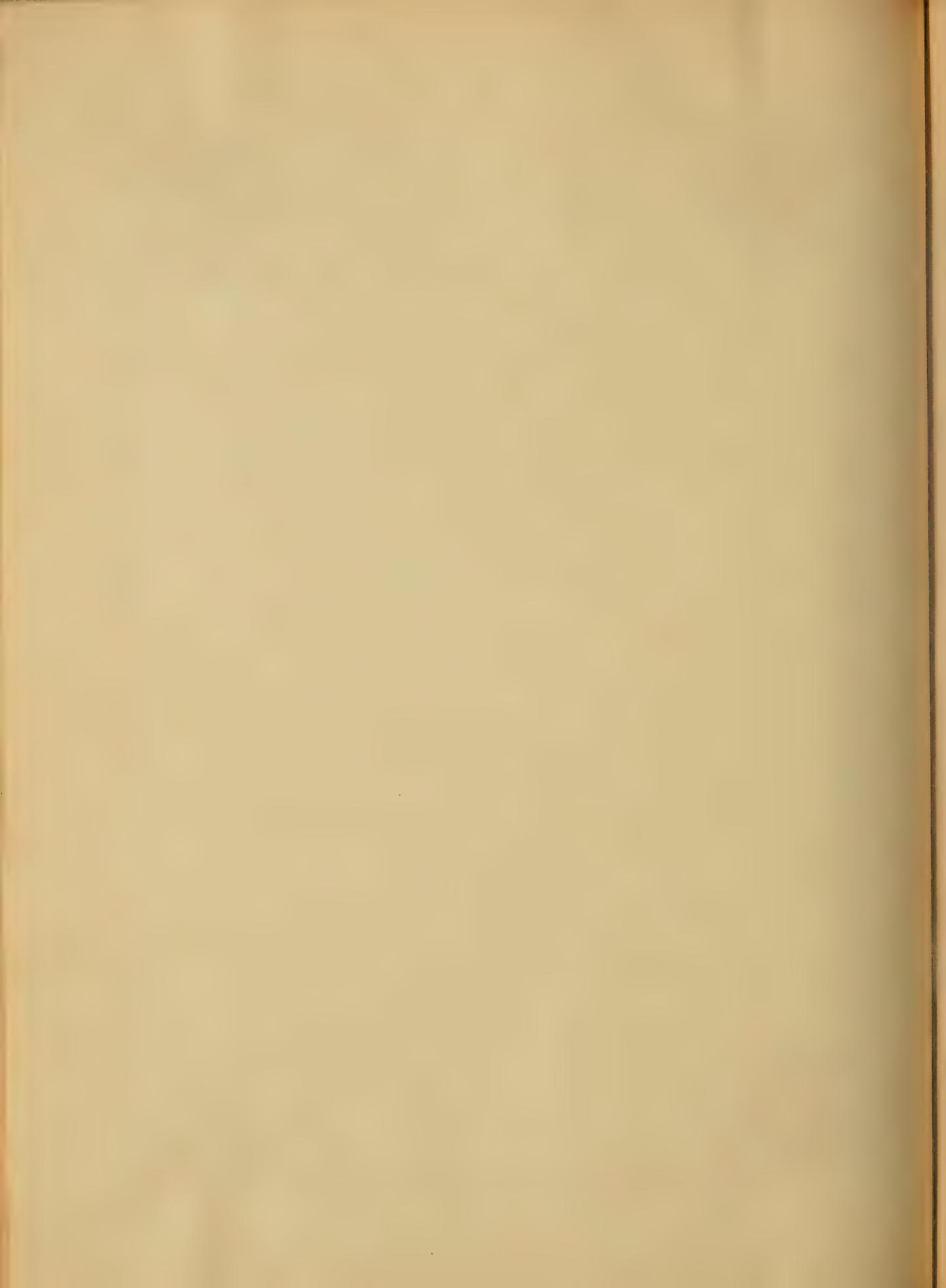
Calibration of Indicator Spring No 3071.

Pres.(lbs)	Compression(in)
89	1.51
80	1.34
70	1.17
60	.99
50	.80
40	.64
30	.45
20	.27
10	.09
Atm.	0

Ascending Readings.

Pres.	Comp.
85	1.44
80	1.35
70	1.19
60	1.00
50	.82
40	.64
30	.44
20	.22
10	.06
Atm.	0

Descending Readings.



Calibration Curve

of

Gauge 1899434.

I - Ascending Curve

II - Descending

Scale 1 in = 16 lbs.

120

100

80

70

60

50

40

30

20

10

0

True Readings.

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Gauge Readings.

Calibration Curves

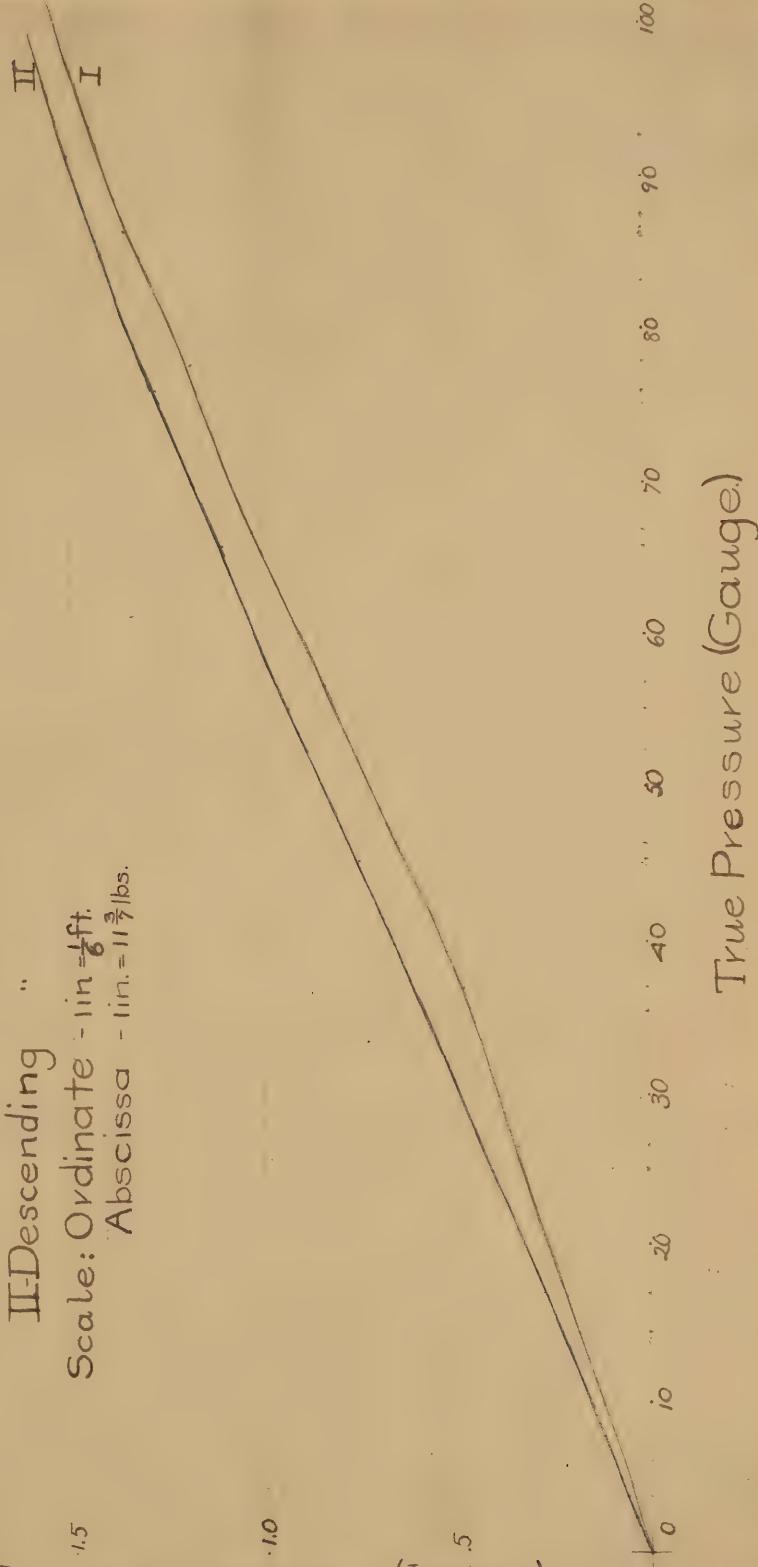
of
Indicator Spring No. 3257 (60 lbs.)

I-Ascending Curve.

II-Descending

Scale: Ordinate - in $\frac{1}{6}$ ft.
Abscissa - in. = $11\frac{3}{4}$ lbs.

Spring Compression (Inches)



Calibration Curves

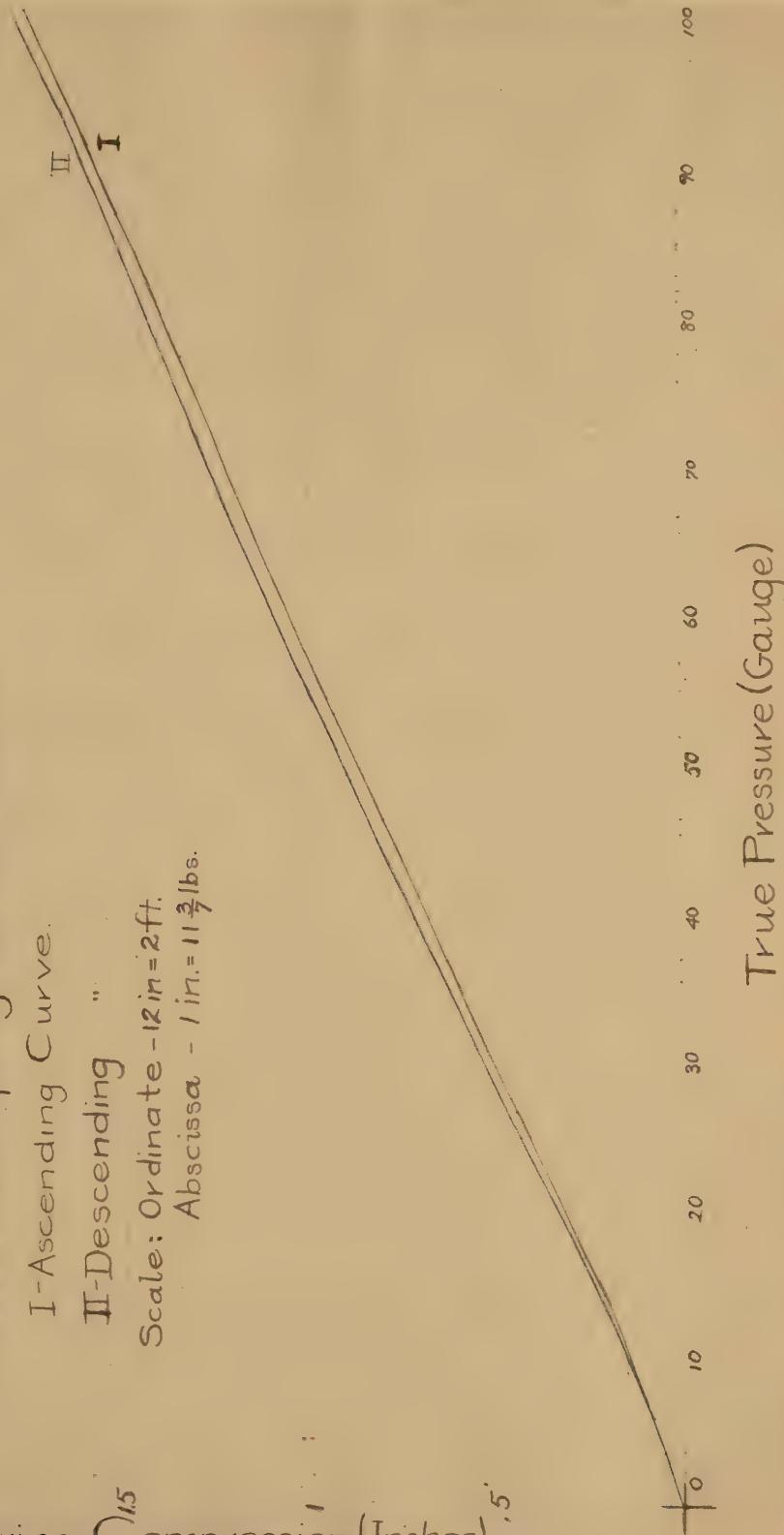
of
Indicator Spring No. 3071 (60 lbs)

I-Ascending Curve.

II-Descending "

Scale: Ordinate - 12 in = 2 ft.
Abscissa - 1 in. = $11\frac{3}{7}$ lbs.

Spring Compression (Inches) $^{1.5}$



3-The engine used to make these tests is a horizontal, single expansion machine of the "Porter-Albin" type, made by the "Southwark Foundry and Machine Co." of Philadelphia.

It is rated at 45 horse power with $\frac{1}{4}$ cut off and 280 rev. per minute. The speed has, however, been geared down to about one half of that at which it is rated. For a complete description of this engine and its parts I would refer to the thesis by "Witmer" and "Rudis" of '900" class.

The clearance volume of this engine was found by placing the engine on its dead centre and then pouring in water until this

clearance volume was filled, noting the time taken to pour and the amount of water poured. Of course some of this water leaks out and this must be corrected for. This is done in the following manner: the time at which the clearance volume is full of water is noted; the water is then allowed to leak for a short while; then more water is poured in until the clearance volume is again full, the time being again noted. By knowing the amount of this extra water poured in, we thus know how much has leaked out between the times at which the clearance volume is full, and from this can get the rate of leakage which is used to correct our first

result. Then when we have found the corrected quantity of water taken to fill the clearance volume it is very simple to find this volume in cubic feet.

From several tests made in the above manner the following mean results were obtained.

Head and clearance = .0308 cu. ft.

Crank " " " = .0316 " "

Below are given the other constants of the engine.

Inside cylinder diameter = $8\frac{1}{2}$ "

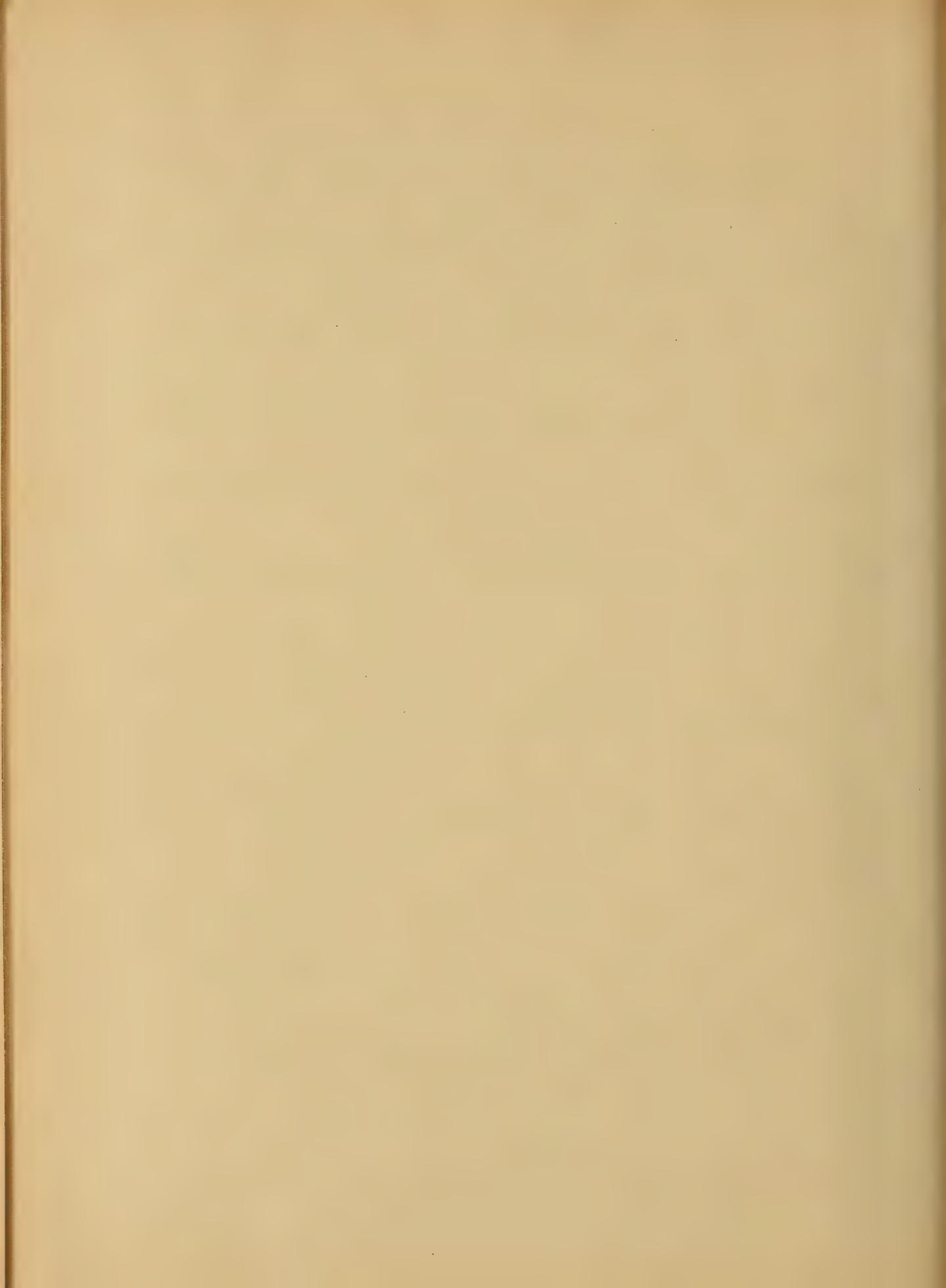
Piston rod " " = $1\frac{1}{2}$ "

Area head and of piston = $53.64\frac{1}{4}$ "

" crank " " " = $48.88\frac{1}{4}$ "

Length of stroke = 16"

Length of connecting rod = 48"



4-In working up the results of this thesis, Peabody's tables of "Saturated Steam" were very extensively used. The following is the notation used in these tables and in this thesis.

p = pressure, pounds per square inch, absolute.

t = temperature, degrees, Fahrenheit.

γ = heat of liquid, B.T.U.

λ = total heat in ".

η = heat of vaporization.

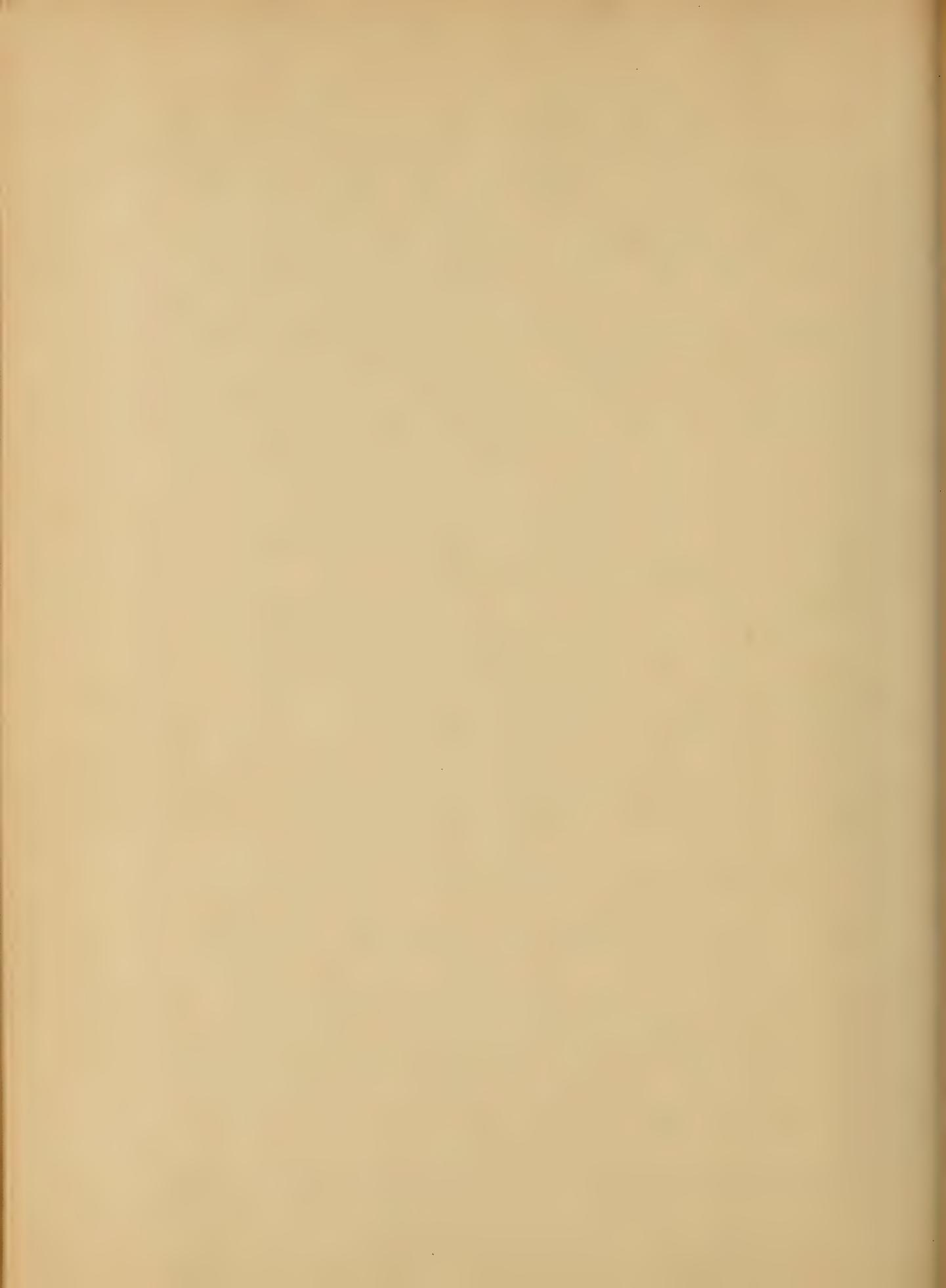
ρ = " " internal work.

$\Delta p u$ = " " external ".

$\int \frac{c dt}{T}$ = entropy of liquid.

s = specific volume of steam.

r = weight in pounds of 1 cu. ft. of steam.



It would also probably be well to refer to "Spangli's Notes on Thermodynamics" for many of the principles of heat which are used here.

5- The following test results were calculated in the manner given below.

Revolutions per minute (R.P.M.) is the number of revolutions recorded in a given time, divided by that given time in minutes.

Quality of boiler steam (x_1) is found by the formula on page 15.

Quality of exhaust steam (x_2), by formula on page 21.

Steam rate per minute is found by dividing the weight of condensed steam discharged from the condenser in a given time, by that given time in minutes.

Steam rate per revolution is the steam rate per minute divided by the rev. per minute.

The amount of steam supplied to the engine per revolution is assumed to be divided between the two ends of the cylinder proportionally to the areas of the corrected indicator cards.

Pressure of supplied and exhaust steams is found by correcting the pressures read by the respective calorimeter gauges.

Mean Effective Pressure (M.E.P.)

the cards being corrected for spring pressure, is found by the following relation or formula:

$$\text{M.E.P.} = \frac{\text{area of card, } \text{sq. in.}}{\text{Length of card, in.}} \times \text{Spring scale.}$$

Indicated horse power (I.H.P.) is found thus:

$$\text{I.H.P.} = \frac{P h (A_h + A_c) N}{33000}, \text{ where}$$

P = mean eff. pressure, M.E.P.

h = stroke of engine, in feet.

A_h = area head end of piston, sq. in.

A_c = " crank " " " " "

N = Rev. per minute = R.P.M.

Brake horse power

(B.H.P.) is the power delivered by the engine and measured by the Prony brake. It is

calculated as follows:

$$B.H.P. = \frac{l \cdot h \cdot 2\pi N}{33000}$$

l = brake arm, in feet, = 106".

h = pressure, in pounds, on
brake scales = brake scales
reading less than zero reading
zero reading with brake
resting on scales, and determined
by the method on page 9, is
equal to 66 pounds.

N = Rev. per minute as
found above.

Mechanical efficiency
is found thus:

$$\text{Mech. Eff.} = \frac{B.H.P.}{I.H.P.}$$

It is necessary for
this thus to know the
amount of strain in the

cylinder during expansion. This is equal to the amount supplied per cycle added to that which is in the cylinder during compression.

The amount of steam in the cylinder during compression is determined in the following manner: from the corrected indicator card, we measure the pressure and volume of the steam in the engine cylinder at the beginning of compression; next, assume that at this point the quality of the steam in the cylinder is the same as that in the exhaust. We thus know the quality, volume, and pressure of this steam which is in the cylinder during

compression. By using Peabody's "Steam Tables" we can easily find the weight of steam in the cylinder during compression, as follows:

$$m = \frac{V}{x \cdot s_p}$$

s_p = specific volume at pressure measured

V = volume measured.

x = dryness of steam.

By means of the formulae given, the following table of "test results" has been calculated. The first table gives the results ordinarily calculated for an engine test.

The second table gives results and data needed to work this thesis.

Table I

	Test I	Test II
Rev. per minute	168	165
Steam supplied per minute, pounds.	28.8	21.2
Mean. Eff. Pres.	40.3	32.5
Ind. horse power.	27.15	21.55
Brake .. .	22.3	16.3
Mechanical Efficiency	82.1%	75.9%
Pounds of steam per I.H.P. hour, dry.	63.3	58.8
Pounds of steam per B.H.P. hour, dry.	77.1	77.7
Pounds of condensing water per minute.	312.6	312.6

Table II.

	Test I	Test II
Absolute boiler pressure.	118.8	115.0
Absolute exhaust pressure.	16.01	16.11
Head end back press. (from card)	20.5	18.2
Crank end back press. " "	20.5	18.2
Value of x in supply pipe	.995	.996
" " " " exhaust	.974	.953
Area corrected card. (Head)	4.78 \square	3.94 \square
" " " " (Crank)	4.89 \square	3.85 \square
Steam per revolution (lbs)	.1713	.1285
Steam to head end supplied (lbs)	.0847	.0649
" " crank " " "	.0866	.0636
Steam in Head end during compression	.00612	.00627
" " Crank " " "	.00662	.00634
Total steam in Head cycle.	.0908	.0712
" " " Crank "	.0932	.0699

6. Diagrams and explanations (see pages following.)

A, B, E, & F are simple tracings
of the actual indicator cards which
were taken.

A = "head" end card (mean) from test I.

B = "crank" " " " " " I.

E = "head" " " " " " II.

F = "crank" " " " " " II.

The atmospheric pressure line is
shown on all of the diagrams.

C, D, G, & H are these
same diagrams on cards, A, B, E, & F,
with their corners squared. This
is done by simply drawing in
even smooth lines and curves
in place of the somewhat ir-
regular ones of the actual cards.

By means of these square cards, we can show the important points of the stroke distinctly. Having these diagrams, C, D, G, & H, we then draw a series of horizontal lines at the proper height above the "atmosphere" so as to correspond to and represent the pressures as taken from the spring calibration curves on pages 32 and 33. These lines are shown in red, the "ascending" spring curves being used to place them on the compression ends of the cards, and the "descending" spring curves for the expansion ends. This is done thus because the indicator piston and pencil are moving upwards while making the compression ends of a card and downwards while drawing

the expansion curve.

The corrected gauge pressures are marked on the diagrams opposite the proper pressure line.

Calibration curve of spring no. 3257 is always used for cards taken from the head end of the engine cylinder, and curve of spring no. 3091 for cards from the crank end.

A, B, C, & D belong to test I

E, F, G, & H " " " " II

The length of these cards as measured is :

$$\text{mean length} = 4.2"$$

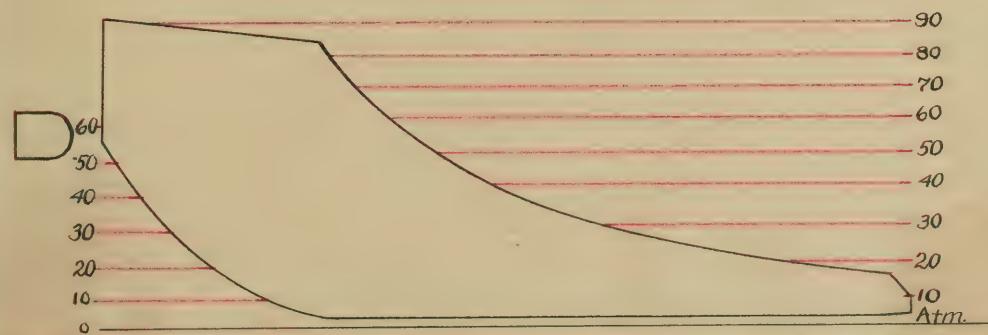
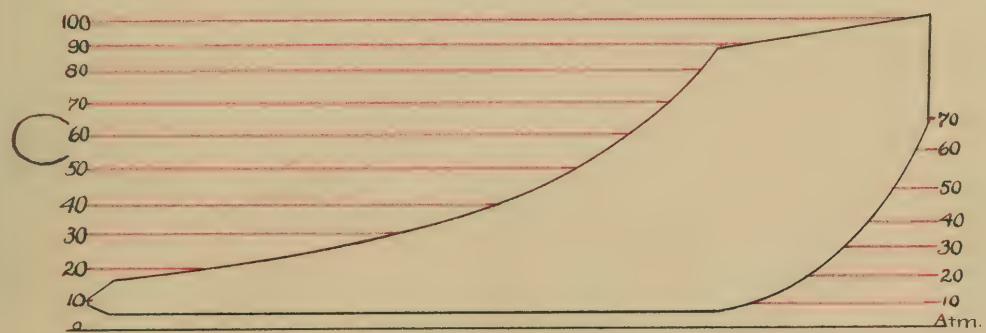
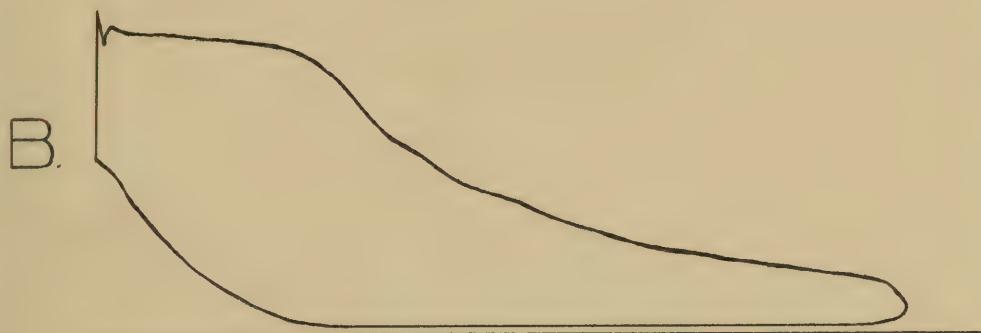
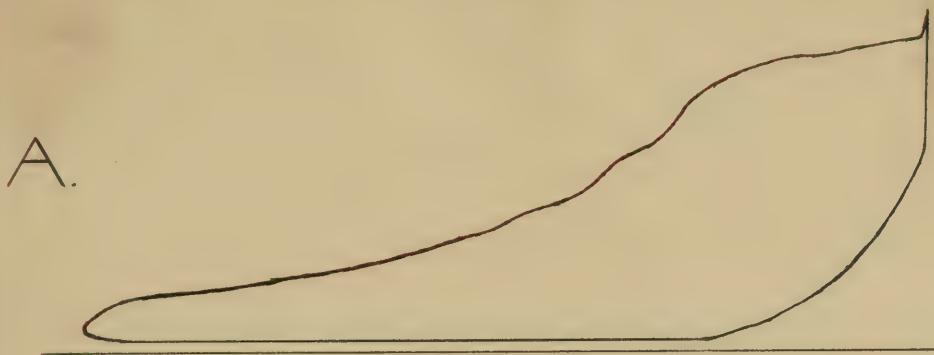
Volume scale of these cards is thus:

for head end cards,

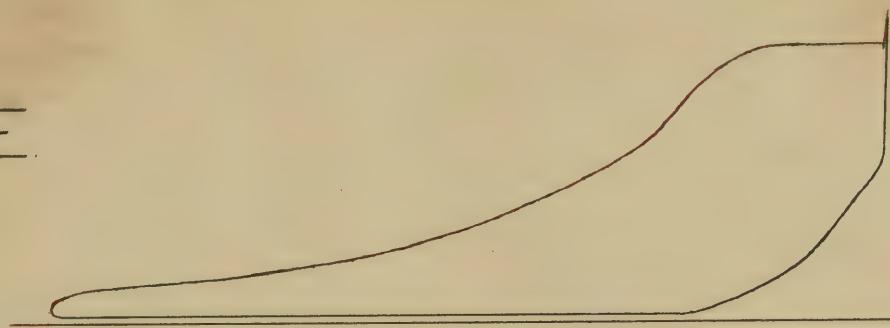
$$1" = \frac{16}{4.2} \times \frac{50.64}{1728} = .01115 \text{ cu. ft.}$$

for crank end cards.

$$1" = \frac{16}{4.2} \times \frac{48.88}{1728} = .01085 \text{ cu. ft.}$$



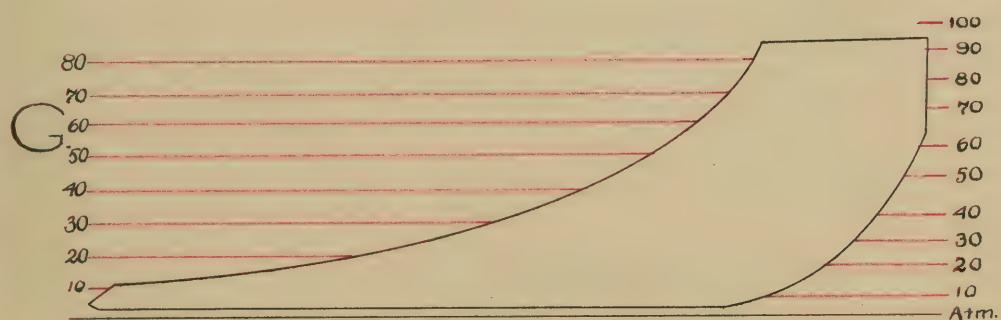
E



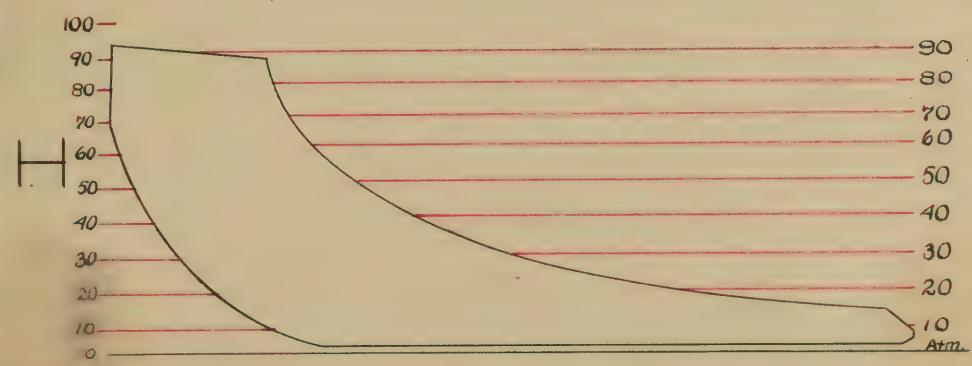
F



G



H



Having now these square
carded cards, C, D, G, H, we proceed
to correct for the variation in
the indicator spring compression.
At the same time that this is done,
however, for convenience we changed
the scale of the cards so as to
have a uniform pressure scale of
50 pounds to 1 inch compression
and a card length of 6 inches.
For these corrected cards, see
diagrams K, L, N, R on pages 58 to 61.

The method of constructing
these corrected cards is as follows
(see diagram K, page 58 for ref-
erence). Diagram K is the
corrected drawing of card C, page 49.
Refining now only to the upper
part of K; lay off a horizontal
distance a-b equal to 6 inches,

this representing the card length or stroke to a scale of:

$$1" = \frac{16}{6} \text{ inches.}$$

At the ends of the stroke ab draw the two vertical lines ac and bf.

Starting now with the line ab as the absolute zero of pressure, lay off on ac the distance ad equal to the atmospheric pressure, 14.7 pounds, to a scale of $1" = 50$ pounds.

Through d draw the horizontal line shown dotted red and marked "Atm". Then using the vertical scale of $1" = 50$ lbs., draw the series of horizontal lines, shown dotted red, to represent each increase of 10 lbs. in the true gauge pressure. We are then ready to construct the diagram K from the diagram C.

The horizontal or volume scale

of the diagrams K, h, N, R is then:

$$1'' = \frac{16}{6} \times \frac{50.64}{1728} = .0780 \text{ cu. ft.}$$

That is, cards from both ends of the engine are reduced to the scale of a head end card, the head end piston area being 50.64 in.^2 .

The volume occupied by the steam at the different pressures on the card C is measured along the horizontal volume lines, to the scale of the diagrams C, P, G, H. Then these volumes are laid down, to the new volume scale, on their respective pressure lines of the diagrams K, h, N, R. Then through the points thus obtained, we draw the corrected indicator cards as shown.

Having now the corrected

indicator cards with the absolute zero pressure line drawn below. we make a series of horizontal lines, shown full red, to represent each increase of 10 lbs. in the absolute pressure.

Then lay off the horizontal distance ao equal to the clearance volume, to the new volume scale, $1" = .078 \text{ cu. ft.}$ Draw the vertical line oe, this representing the line of zero volume. Now we have from the table on page 45, the total weight of steam, M , in the cylinder during expansion. Then by means of the Steam Tables, we can find for each absolute pressure, the volume which would be occupied by this

quantity M , if it were all dry straw, i. e. $x = 1$. Thus we are able to get points through which to draw the curve \ln , which is the expansion curve of dry straw. Then having the expansion curve \ln , of M pounds dry straw, and also the full black line rs , which is the actual expansion curve of M pounds of straw; then at any pressure P , the ratio of the volume \underline{hP} to the volume \underline{hV} is the value of x at the point P , i. e. the quality of the straw at that point of the stroke.

Below each of these corrected cards in the diagrams K, L, N, R , is drawn a curve sharing the

the variation of this quantity x during the expansion part of the stroke. The abscissæ of this curve show points in the stroke and the ordinates, the value of x . The value of the ordinates of this curve are shown by the horizontal red lines drawn with the corresponding value of x marked thereon. Scale is: 1" ordinate = .1 in value of x .

From knowing as we do, the weight of steam M in each end of the cylinder during expansion and also the pressure and quality of this steam at any point of the stroke during expansion, we are able by means of the Steam Tables to calculate the quantity of heat,

above 32°F , actually in the
strain of each end of the cylinder
during any part of the
stroke between cut-off and
release.

Diagrams are as follows:

K = C corrected, with additions as above.

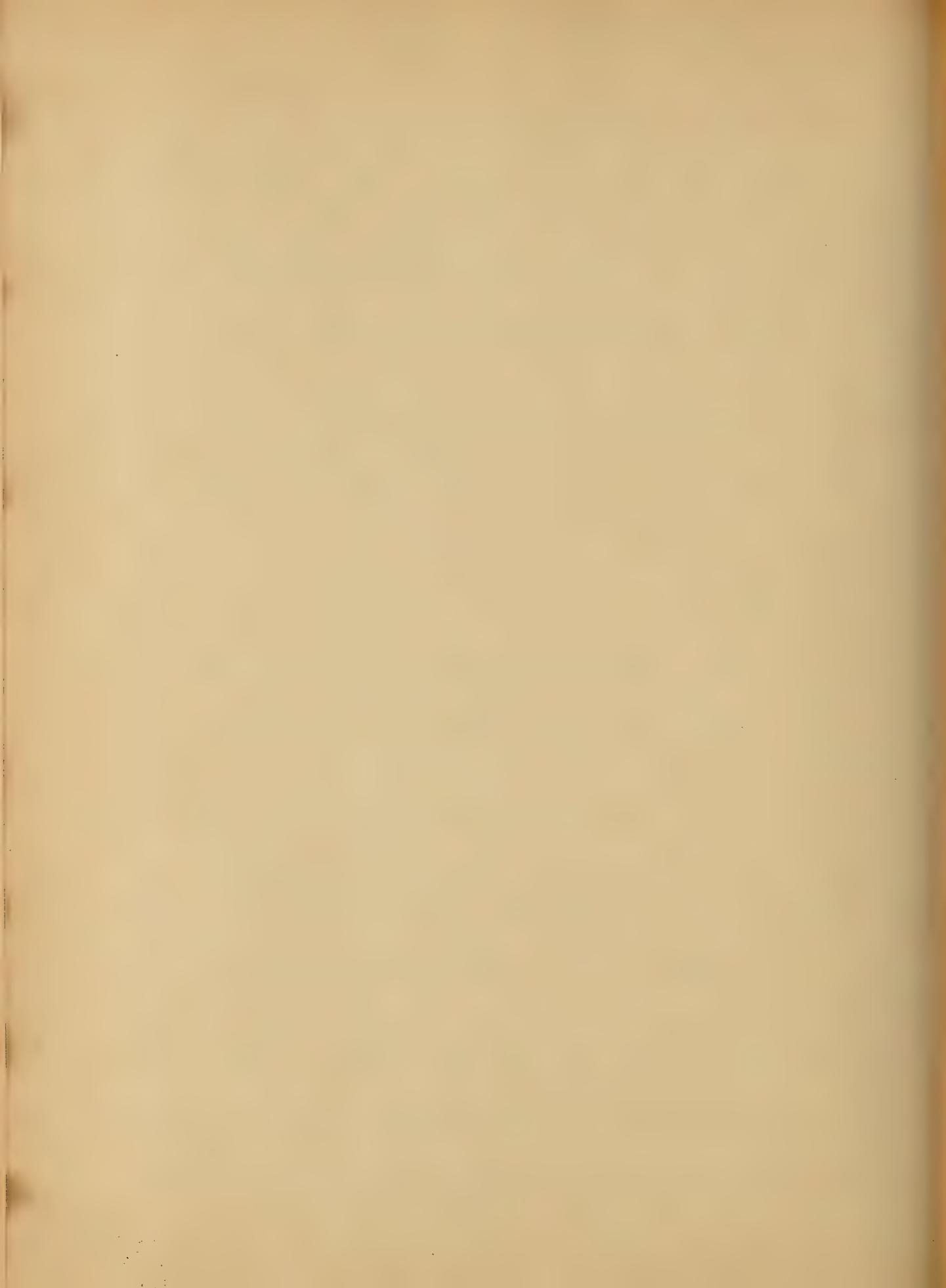
L = D " " " "

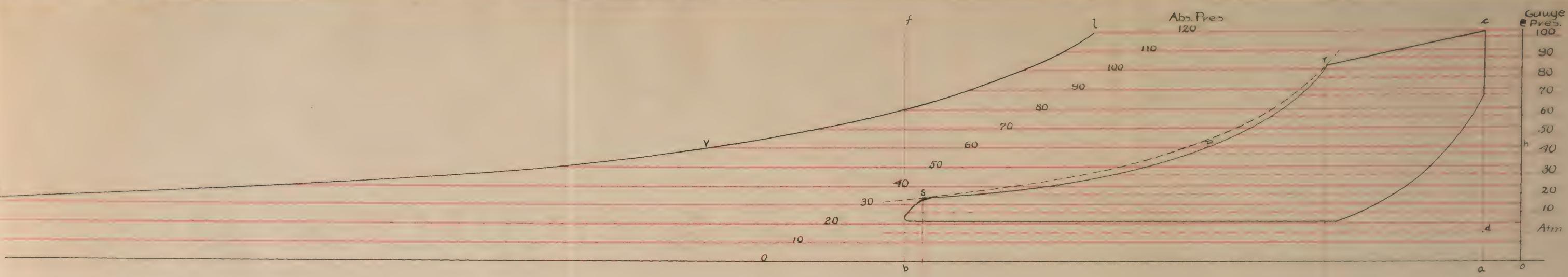
M = G " " " "

R = H " " " "

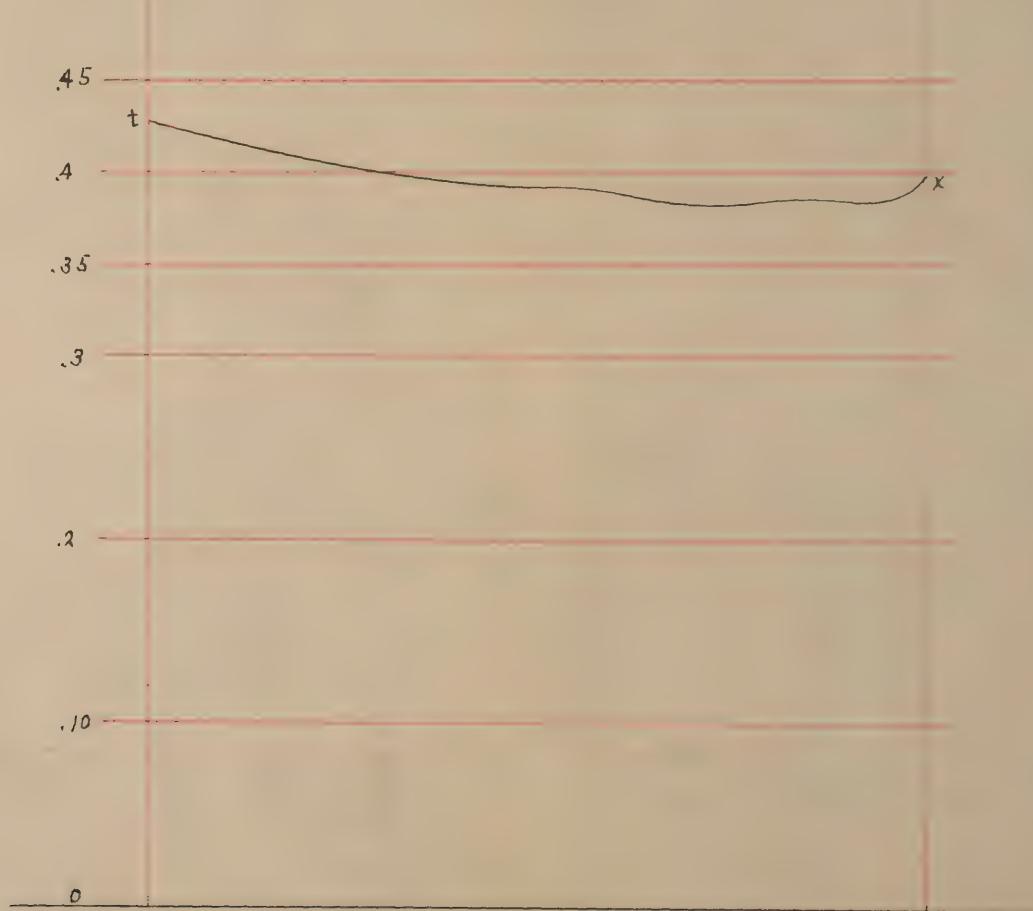
The dotted black curve RS is
the rectangular hyperbola drawn
through point R.

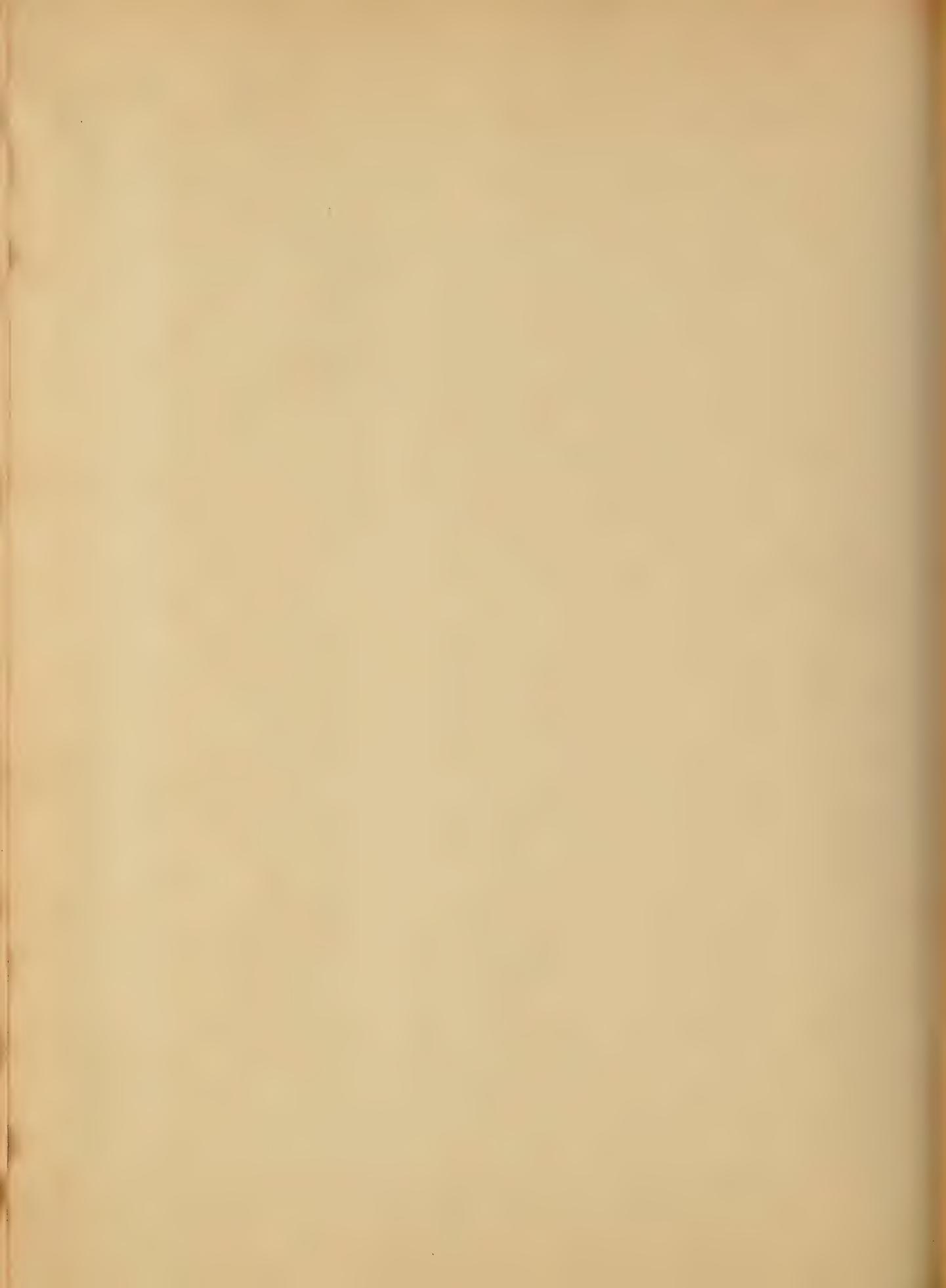
The rise in the x curve of
diagram R, towards the end of the
stroke, is probably incorrect, it
being due to a leaky strain valve.
This is probable since the expansion
curve rises so much above the
rectangular hyperbola.

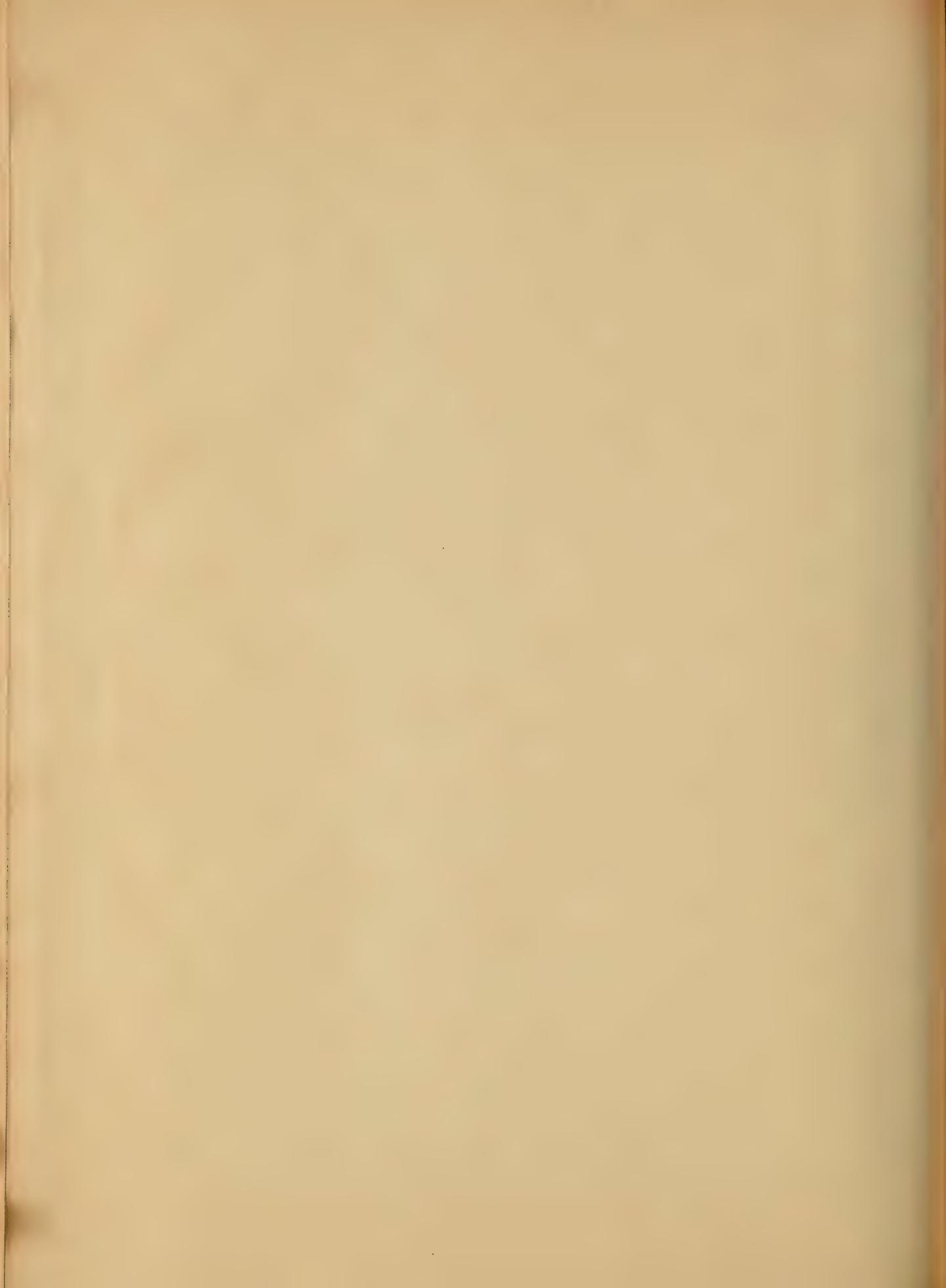




K.

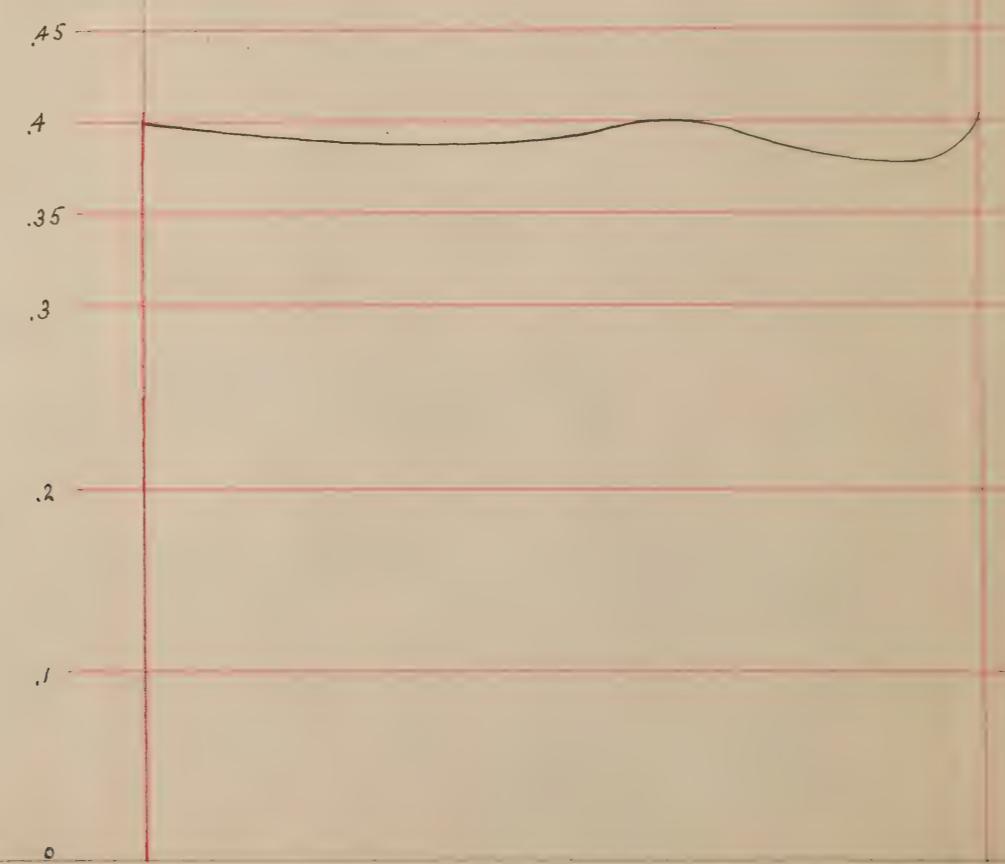
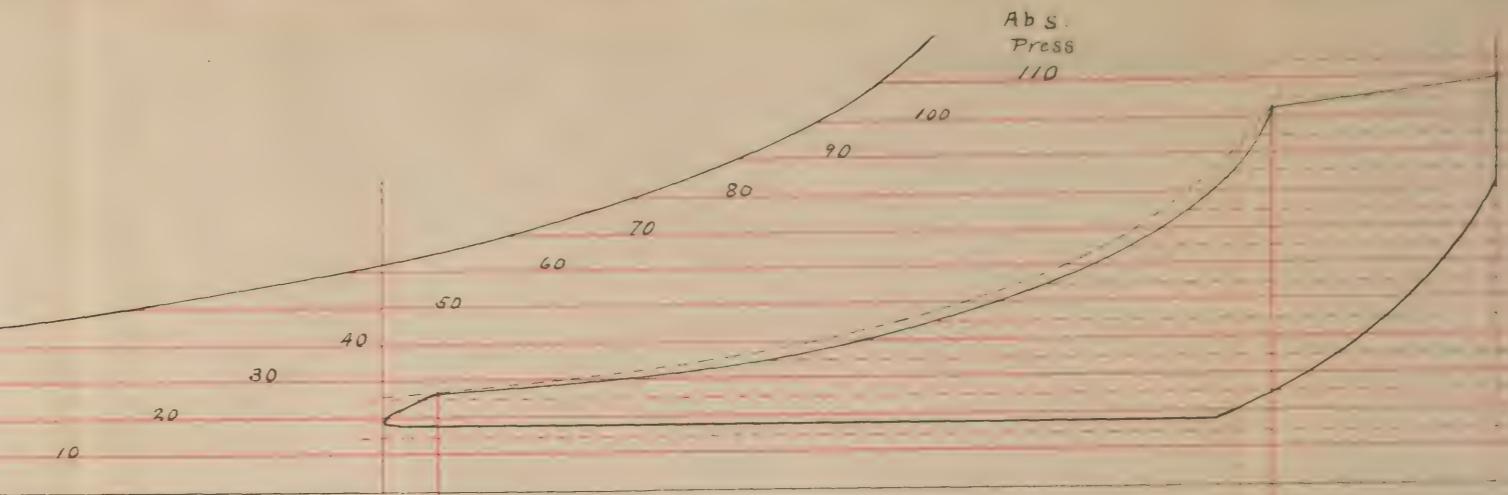




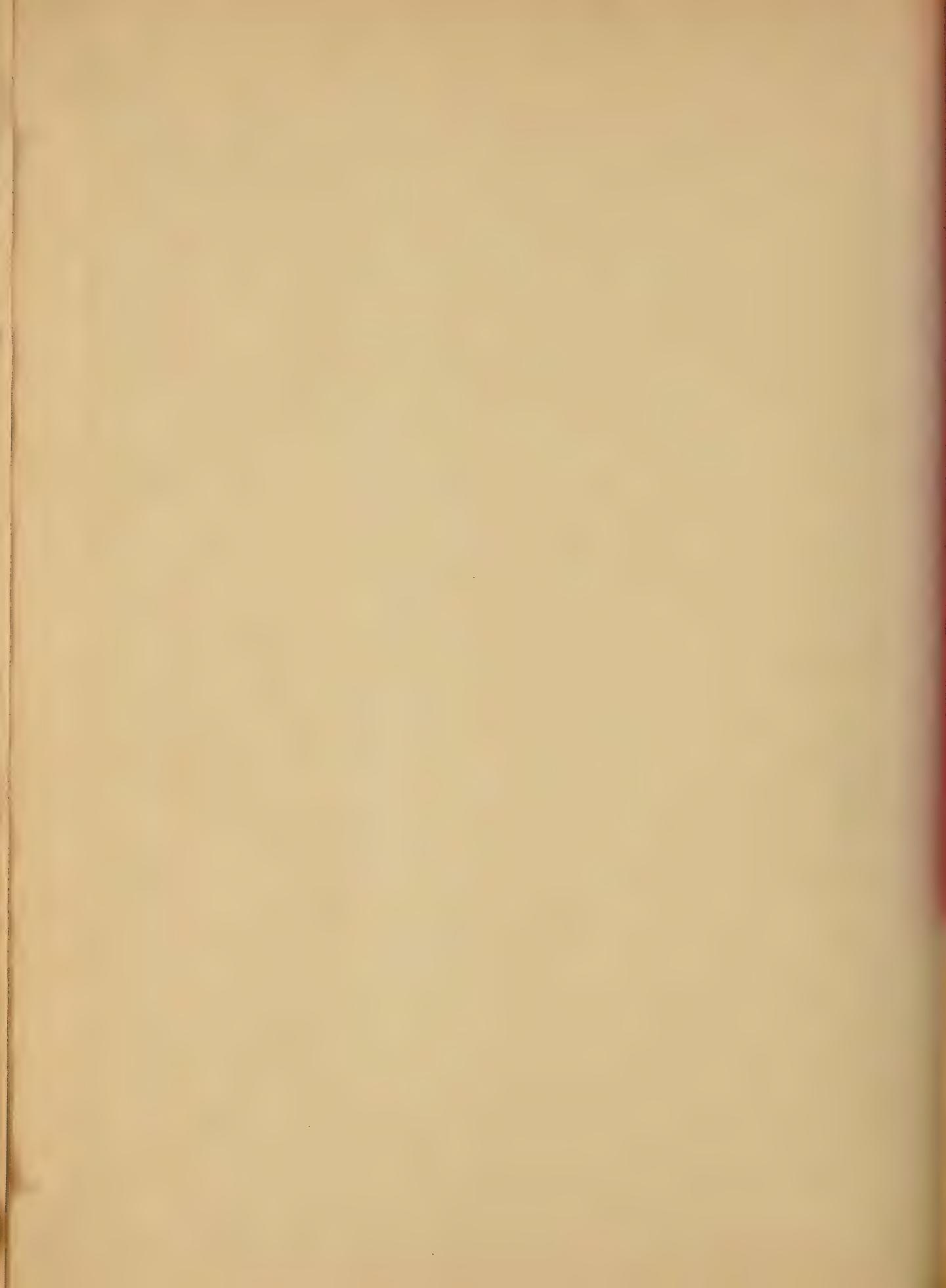


Gauge
Press

100
90
80
70
60
50
40
30
20
10
Atm



N



Length
Tress

100
90
80
70
60
50
40
30
20
10
0
cm

Abs Press

100

90

80

70

60

50

40

30

20

10

0

45

4

.35

.3

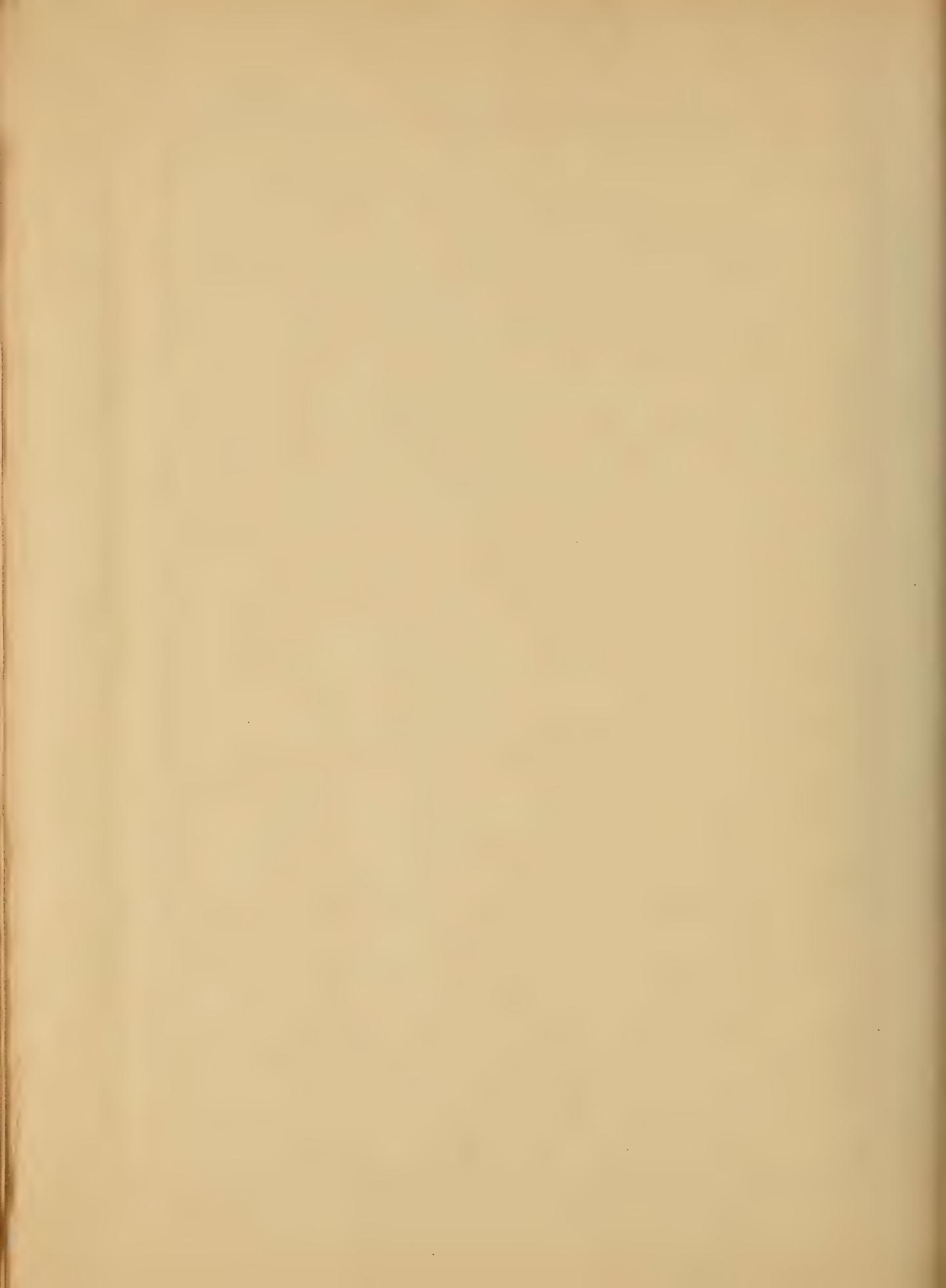
.25

.2

.1

0

R.



7- As we wish to show the rate of heat changes in the steam engine, it is necessary to draw curves between heat and time.

In the curves here drawn, the abscissæ represent time and the ordinates, British Thermal Units. In order to change from points along the stroke to a time axis, we first assume that the engine turns at a uniform rate during the whole revolution.

In drawing these time heat curves, only the first test (I) is "worked up" and its curves constructed. Test II was used to help construct the time-heat curves for test I.

The engine, in this test, ran at the mean rate of



168 R.P.M., with a cut-off of about $\frac{1}{4}$. There are 360° in one revolution, therefore the number of degrees through which the crank turns in one second is

$$\frac{168 \times 360}{60} = 1008.0 \text{ degrees.}$$

In .01 second, then the engine crank turns through $10^\circ 4'$.

Referring now to the diagram T on page 66, suppose we have the crank circle drawn to a given scale. If the scale is $1" = \frac{1}{6}$ inches, and the crank circle is separated into two parts in order to give clearance to the diagram. Knowing that the crank turns through $10^\circ 4'$ in .01 second of time, lay off points on the crank circle

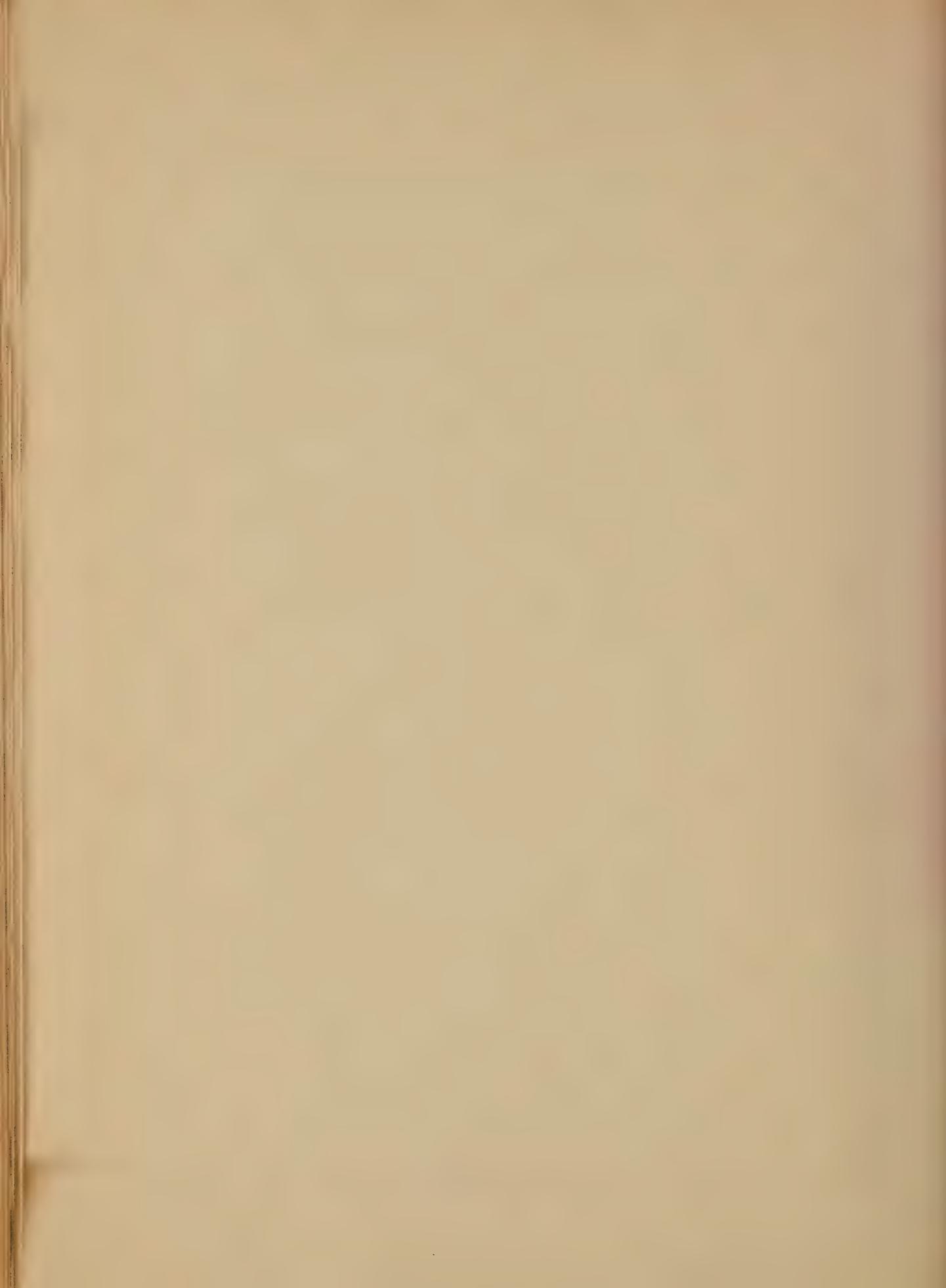
corresponding to every .01 second, starting with the engine on its "dead" center at a zero time.

From the point at which the crank is at any given time, with a radius equal to the connecting rod length, draw an arc cutting the stroke line. The point thus found on the stroke line is the piston position corresponding to the given ~~given~~ time with which we started. By this method points are found on the stroke, corresponding to every .01 second. On the other hand, starting with any point in the stroke we can walk backwards and get the point on the crank circle and also the time corresponding.

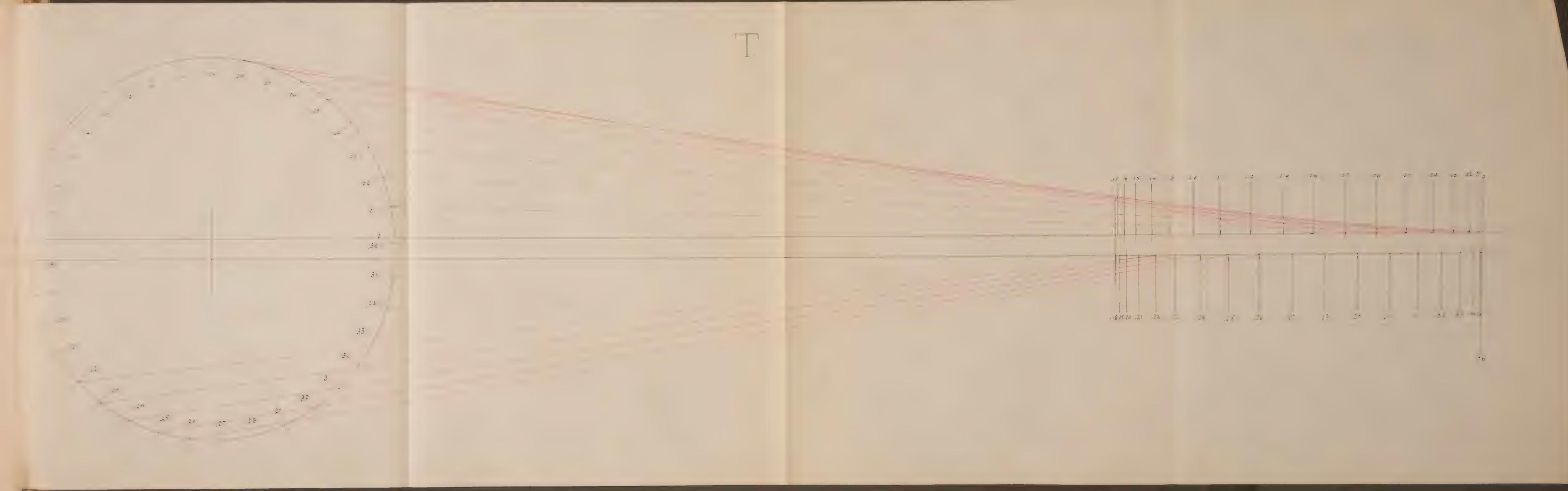


We then lay off a horizontal time axis on which $1'' = .02$ seconds, calling the point of lead and admission (i. e. the "lead" dead-centre) the point of zero time, i. e. the origin from which we start.

An inspection of diagram T will make clear this method as above explained.



T

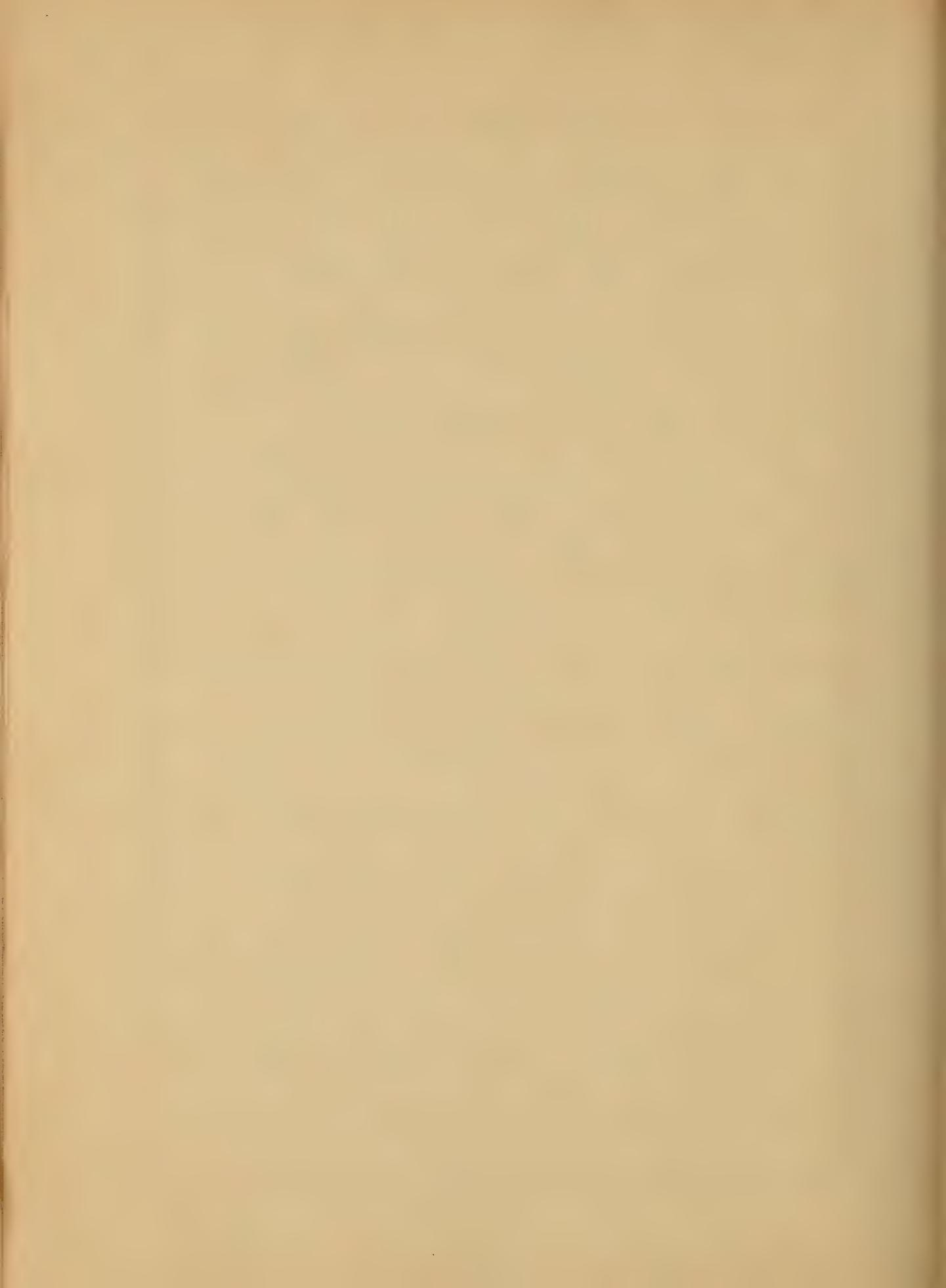




8- Diagrams W and Y represent the variation of heat with time, page 79. Both have the same scale and zero point of time but the zero points for heat are not the same in both even though the heat scale is.

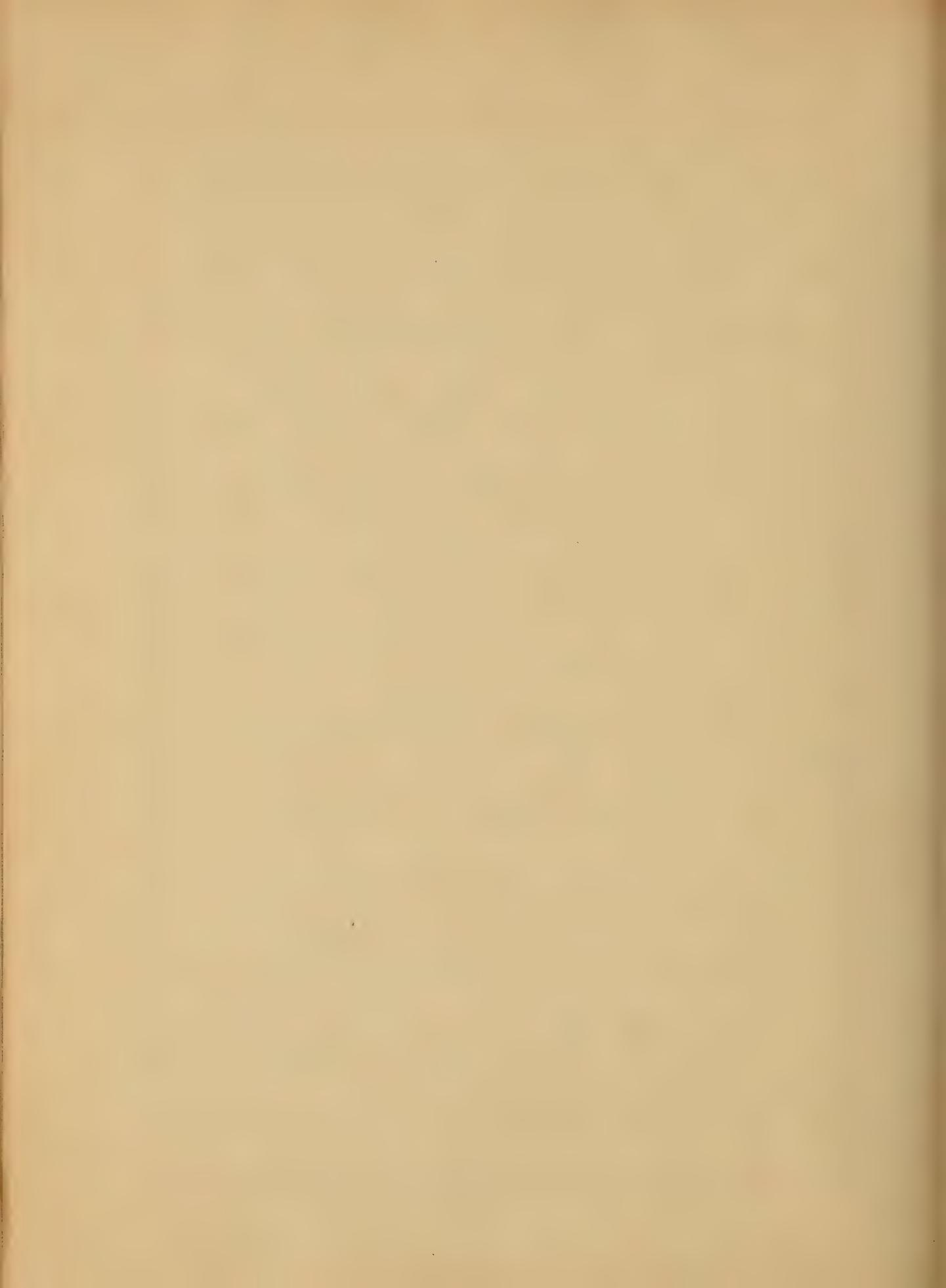
Diagram Y shows the variation of heat held by the steam in the cylinder. Curve a b c d e a is for the "head" end steam and is shown in black ink. Curve f g h i k f is for the "crank" end and is shown in blue. These curves are both gotten in the same manner and by the same method so only one explanation is necessary.

First a horizontal time axis is laid off, the scale line being $1'' = .02$ seconds. We suppose that the point of "head" admission, i. e. head



dead centre is the zero or starting point of time. Then from the stroke diagrams, as on pages 58 and 59, by means of the method explained in paragraph 7, we can put down on this time axis all of the important points of the stroke, cut-off, release, compression, etc. These are shown by the "dotted red" lines and are marked on the diagram.

We know from table II, page 45, the weight of steam in each end of the cylinder during expansion, and also from diagrams K and L, the pressure and quality of this steam at any point of the stroke between cut-off and release. Then by means of the Steam Tables and the simple formula of heat which



is: $H = m(q_p + x_p p_p)$, we have the amount of heat, above 32°F , actually in this steam at any point of the stroke between the cut-off and release. Hence we can construct to any suitable scale the curve bc showing the variation of heat in the steam during expansion. The heat scale line used is $1'' = 10 \text{ B.T.U.}$, the zero line or being the heat at 32°F .

Knowing the weight of steam in the cylinder during compression, as in table II, we suppose that right at the point of compression this steam is of a quality equal to that of the exhaust and of a pressure equal to the "back

"pressure" measured from the card. We can then calculate the heat held by the steam at this point, thus getting the point \underline{e} . By measuring from the card the work of compression we get the heat added during compression and thus get the curve from \underline{e} to \underline{a} . This of course assumes that the compression is adiabatic.

Now we see that the curve rises from \underline{a} to \underline{b} and drops off from \underline{c} to \underline{e} . Let us make a test (II) having admission at \underline{o} and cut-off at $\underline{n'}$. We can then get weight of steam w which has come in the cylinder from \underline{o} to $\underline{n'}$, and also its quality at cut-off, n' .

Now we assume that in test I,

the weight and quality of the steam in the cylinder at the time x_1' are the same as at this time (x') in test II. Thus by making a series of tests II, III, IV, having cut-off points earlier than in test I, we are able to get the points m, m', s and are then able to draw the curve $a-b$.

Very little can be fairly assumed or found out about the curve from c to e . If however the steam expands fully from the point d when to the end of the stroke, knowing the volume and pressure at the end of the stroke, we can calculate for a point d . This assumption of "free expansion" is really a fair one to make for the

following reasons: we see that during the latter part of expansion, that is along the curve b-c, the heat held by the steam is nearly constant, i. e. the cylinder walls are not suddenly giving up any large amount of heat; can we not then suppose that this amount of heat held by the steam would have still remained nearly constant if release had been right at the end of the stroke. Therefore, knowing the quality and pressure at release and also the pressure at the end of the end of the stroke, we can get the quality at this last point, assuming, as it were, "free expansion" and neglecting the work, which

is very small. The curve from \underline{d} to \underline{e} is then a mere assumption or guess. It is simply a continuation of $\underline{c}\underline{d}$ which is curved around so as to meet $\underline{e}\underline{a}$.

We now wish to explain the upper diagram W , on which there are five curves. Here let the horizontal line $o\,p$ represent the amount of heat, above some zero point, which is in the whole engine at the point of zero time, i. e. the lead end admission. Curve σ, x, t, u, v, q, j , shows the amount of heat having entered the engine since the zero time at o . It is gotten as follows: by means of

the "steam supplied" and "length of cut-off" from the several tests I, II, III, & IV spoken of above, knowing the pressure and quality of the steam supplied, we can get the points t , z , y , & x , the heights of which represent the amount of heat having entered the engine since the time, σ , of head admission; from cut-off to the next admission, no heat is entering the engine, hence the curve is a horizontal line from t to u ; the curve then rises from crank admission to crank cut-off in a manner similar to that of the head end. This curve shows as was mentioned before, the amount of heat having entered

the the engine since the zero
time, σ . It is shown in
black on the diagram W.

The blue curve A, σ , B, C, D, E
is to show the amount of heat
leaving the engine in the exhaust.
Knowing the weight and pressure
of the steam leaving the engine
during each exhaust and having
measured its quality by the
Carnotia Calorimeter, we can
calculate the amount of heat
rejected by the engine during
each exhaust. Now from
our "full expansion" idea
in diagram Y the amount of
heat f_l has left the engine
in the exhaust during the
time from t to σ . Then
let $A L$ be equal to f_l and

we have heat leaving the cylinder in the exhaust from A through σ to B. This curve passes through σ since the horizontal through σ is what was taken as the amount of heat actually in the engine at the point of head end admission. This curve from A to B is not necessarily a straight line but is merely taken as such since nothing definite is known about it. From compression at B to the next release at C there is no heat leaving in the exhaust, hence curve from B to C is a horizontal line. We then have a repetition of affairs for the "head" exhaust, as in curve C, D, E.

Starting with the heat having entered the engine in a given time, if we subtract the heat rejected and the work done in that given time, we have left the amount of heat radiated to the outside air. Now we suppose that this radiation takes place uniformly with the time and thus we get a straight line OR for the radiation curve. The ordinates of this curve and also of the curve oWh are laid off below the line oP simply for convenience.

From the indicator cards, we can measure the amount of work done up to any given point. Then plotting

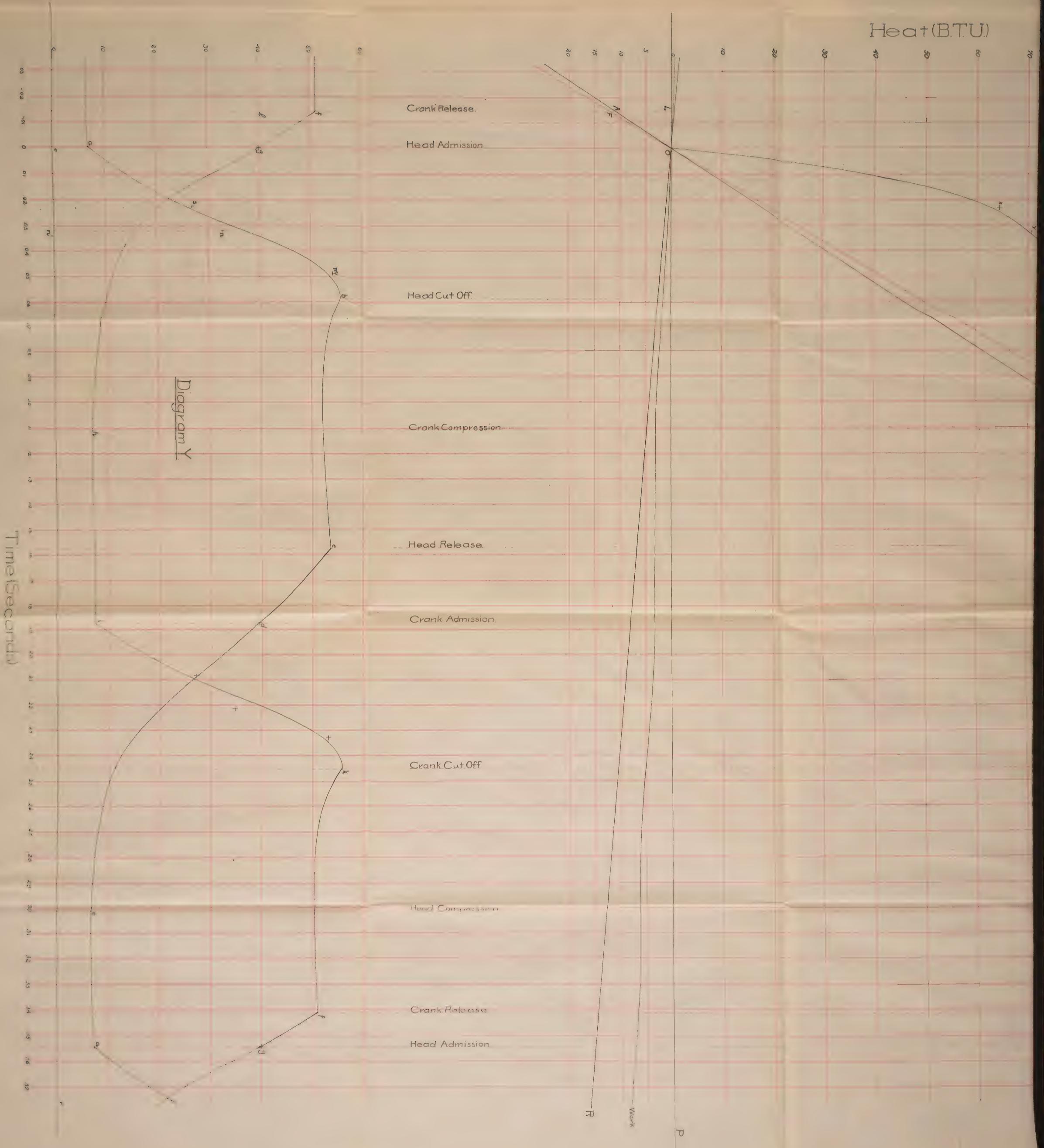
the curve showing work done (in B.T.U) we have, as shown in blue on the diagram W, the curve of work $\sigma \underline{Wk}$.

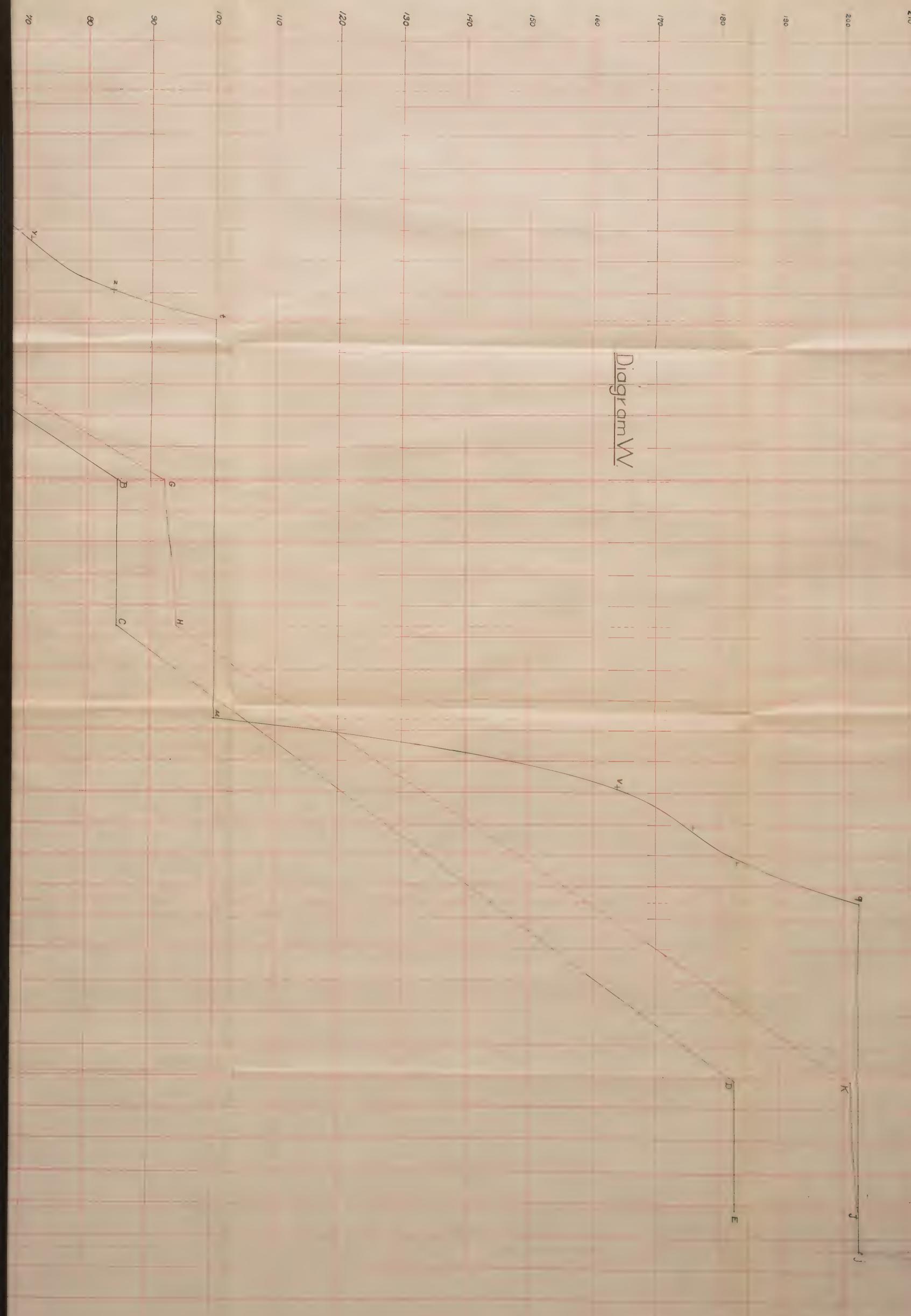
Then adding the ordinates of the curves $\sigma \underline{R}$ and $\sigma \underline{Wk}$ to those of the curve A, B, C, D, E, we have the "brown" curve F, G, H, K, J which shows the amount of heat disappearing, i. e. accounted for, in the engine.

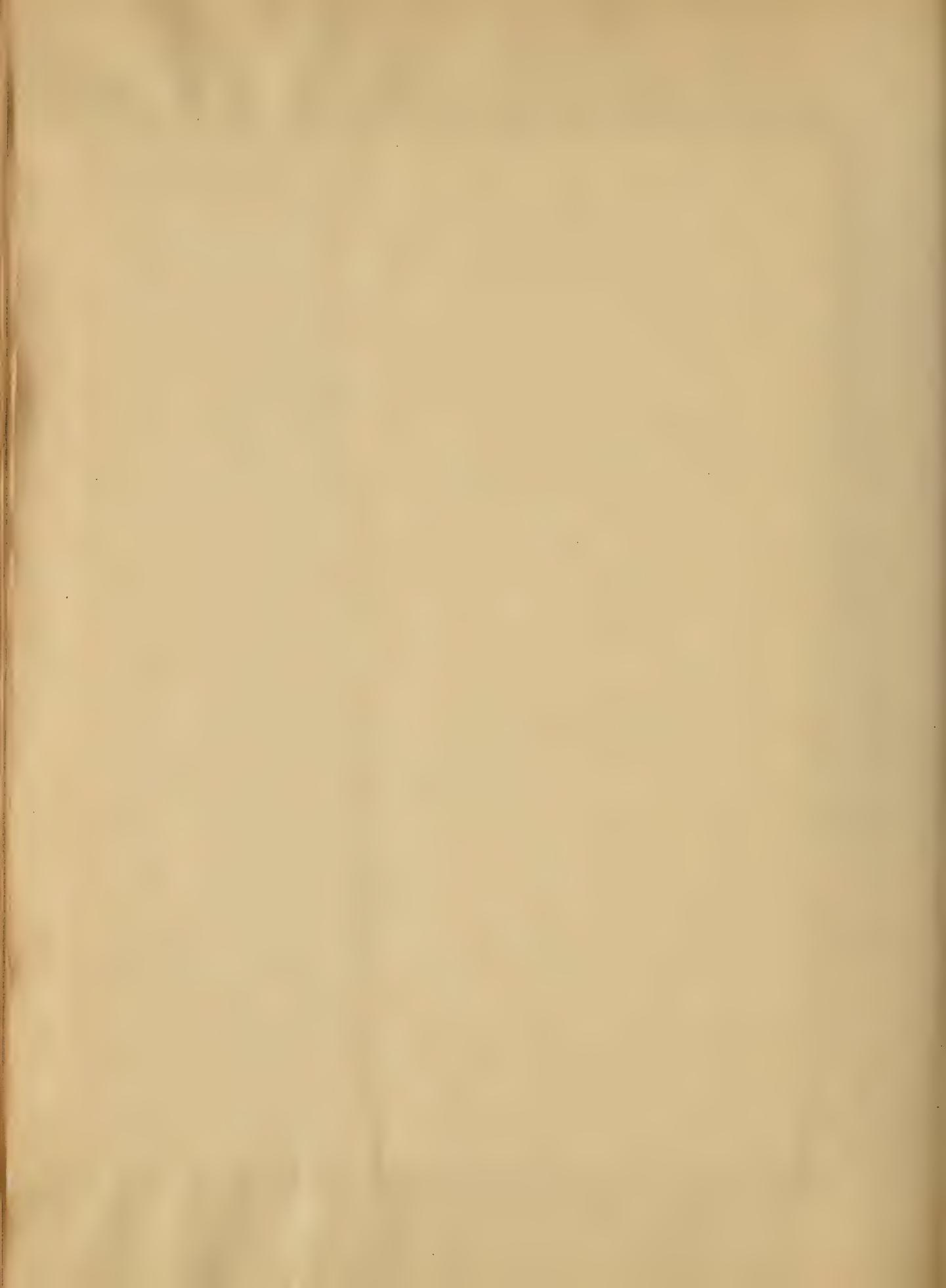
The diagrams W and Y are on page 79.

1" abscissa = .02 seconds.

1" ordinate = 10 B.T.U.



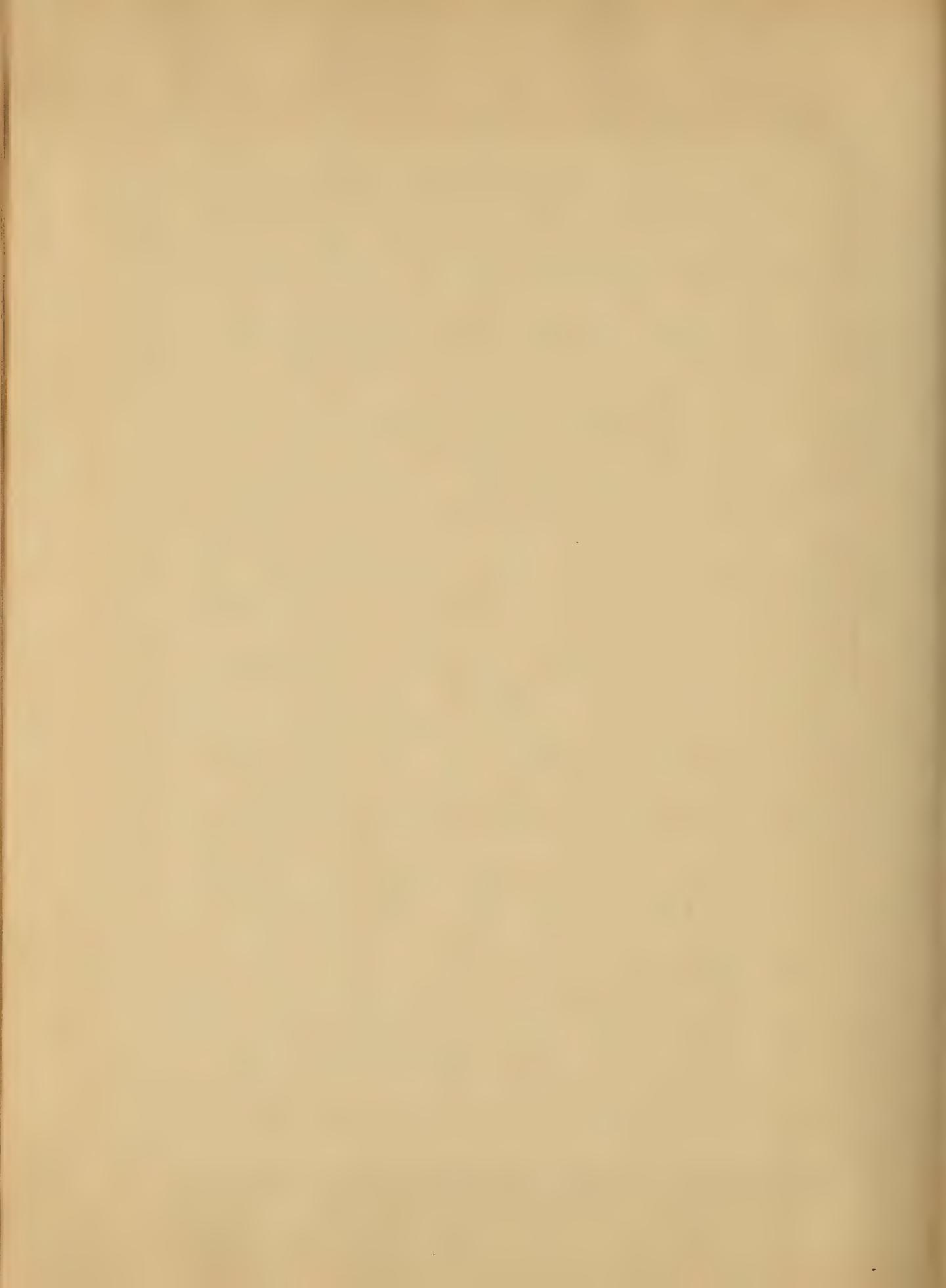




9- In drawing these line-heat curves and in making the calculations for this thesis some assumptions have been made, the fairness of which we now wish to discuss.

In getting the weight of steam in the cylinder during compression, it was assumed that at the point of compression, the quality of the steam in the cylinder was the same as that in the exhaust pipe. A little reflection will show that this is a very fair assumption and even if it were in error, a very small effect would be had on the final results.

The steam supplied to the engine per revolution was assumed to be divided between



the ends of the cylinder proportionally to the areas of the cards. The areas of the cards represent work done and the distribution of strain was thus made proportional to the work done. Perhaps a more fair assumption to make would have been to divide it proportionally to the time during which the admission valves were open, as from the diagram W or Y.

The assumption made in regard to the curve between admission and cut-off as explained on page 70 is as fair a one as could possibly be made under the circumstances.

The assumption of "free expansion" at release is, as will be

seen from the explanation on page 7 a quite fair one.

The assumption of straight lines in the upper part of diagram W is, as was said before, really groundless except for the fact that the points A, S, & B happen to be in a straight line on the "blue" curve drawn.

That the radiation takes place uniformly with the time is without proof except for what we know of radiation in general.

There are doubtless many small errors in the curves drawn, due to graphical methods being employed.

10- From an inspection of the diagrams W and Y on page 79, we see that the heat ordinates of the work and radiation curves are very small as compared with those of the other curves. However, the steam consumption of this engine, as seen from the table on page 44, quite abnormal; and the ratio of the work curve ordinates to those of the other curves would ordinarily be somewhat materially larger.

Nevertheless we see from diagram Y that the total amount of heat held by the steam in both ends of the cylinder remains quite nearly constant, i. e. it is not subject to any

very great fluctuations. Since it is reasonable to suppose that the cylinder walls themselves are not subject to any very great and quick variations in temperature, does it not follow that the total amount of heat contained by the cylinder walls themselves is a quantity which remains nearly constant also. This would seem to lead us to believe that the total amount of heat held in the engine is quite nearly constant. This state of affairs would also seem to be indicated from a combination of the "heat entering" and "heat disappearing" curves shown in diagram W.

That is, the amount of heat held by the engine varies to a not very large extent both above and below a mean value.

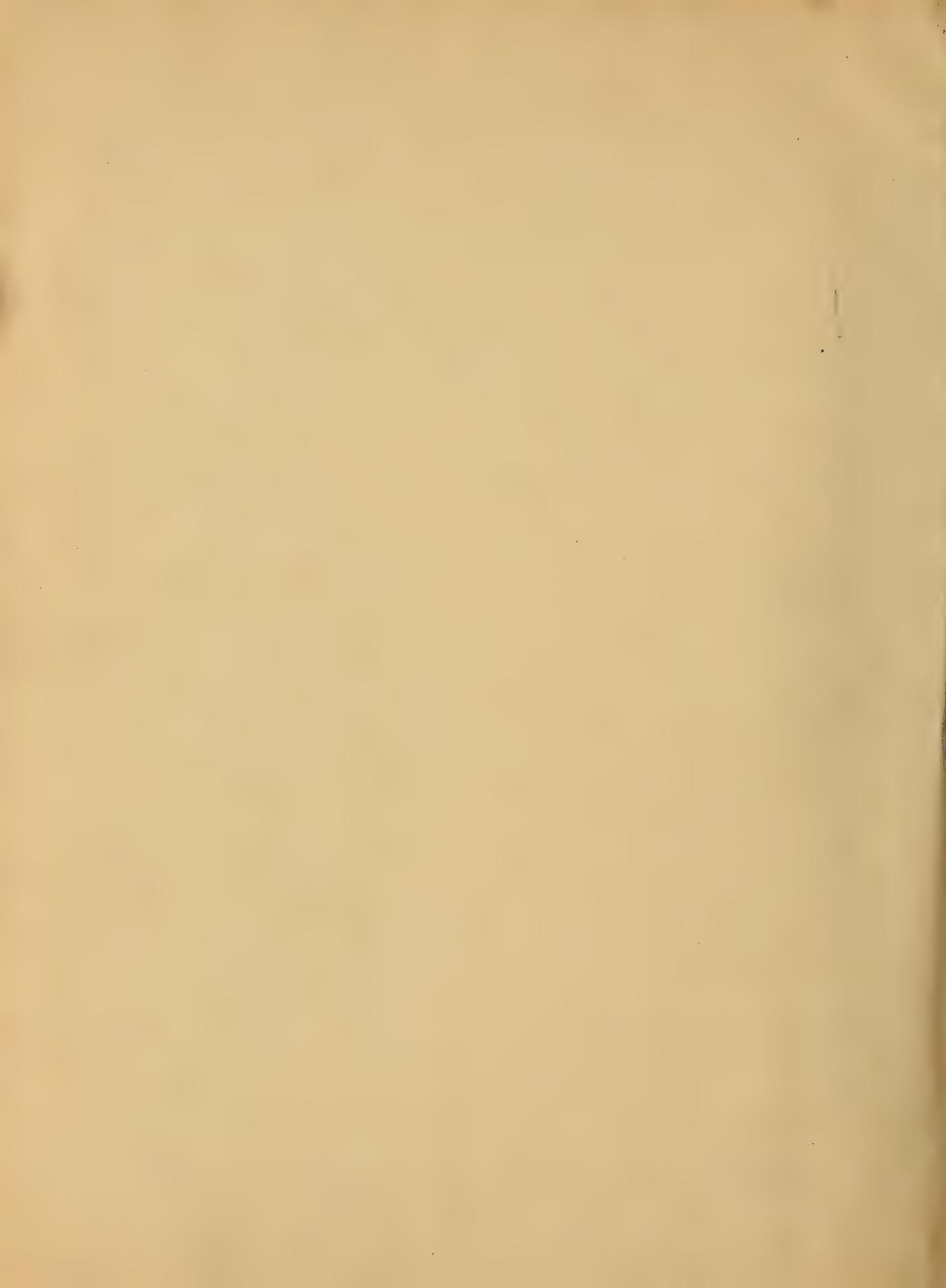
From a comparison of the curves of diagram W a suggestion comes that perhaps the heat entering one end of the cylinder during admission is immediately, by some manner or means, conducted or transferred to the stream of the other end and thus goes out in the exhaust. However one thing is very apparent, that is: the exhaust stream becomes heated up somehow. We have here shown that the quality of the stream at release is only about .4 to .5 while in the exhaust pipe it is somewhere

about .96. On expanding
fully from the pressure at
release down to the pressure
of the exhaust, the value of
 χ only rises slightly, such as
from .4 to .45. Therefore,
as was said before, it is very
evident that a large amount
of heat must be added to
the exhaust steam at some-
time between release and
the beginning of compression,
i.e. the time when this
exhaust steam has completely
left the cylinder.

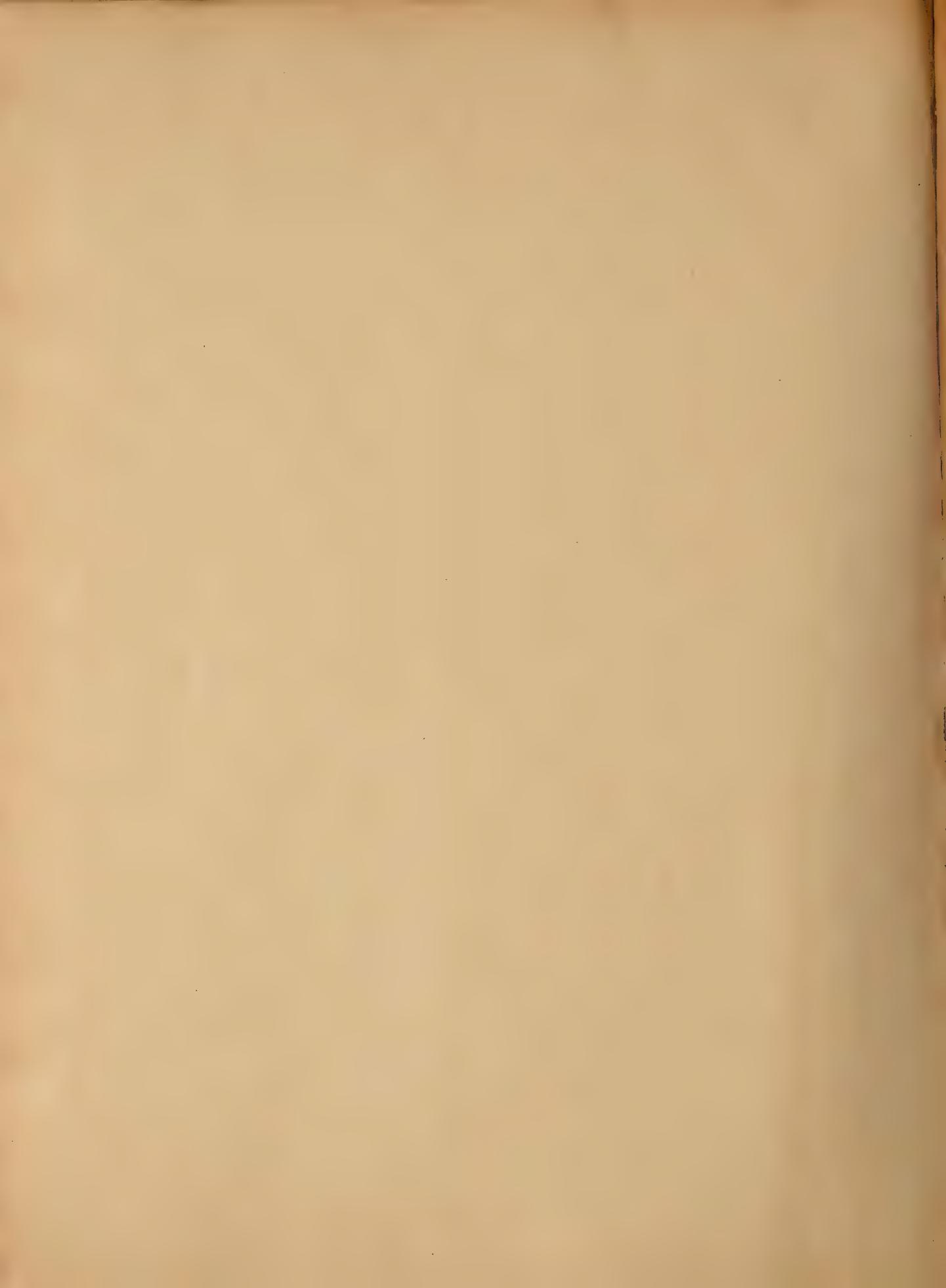
A fuller and more
exhaustive study of this phe-
nomenon along the line here
indicated would, I feel sure,
lead to a solution of

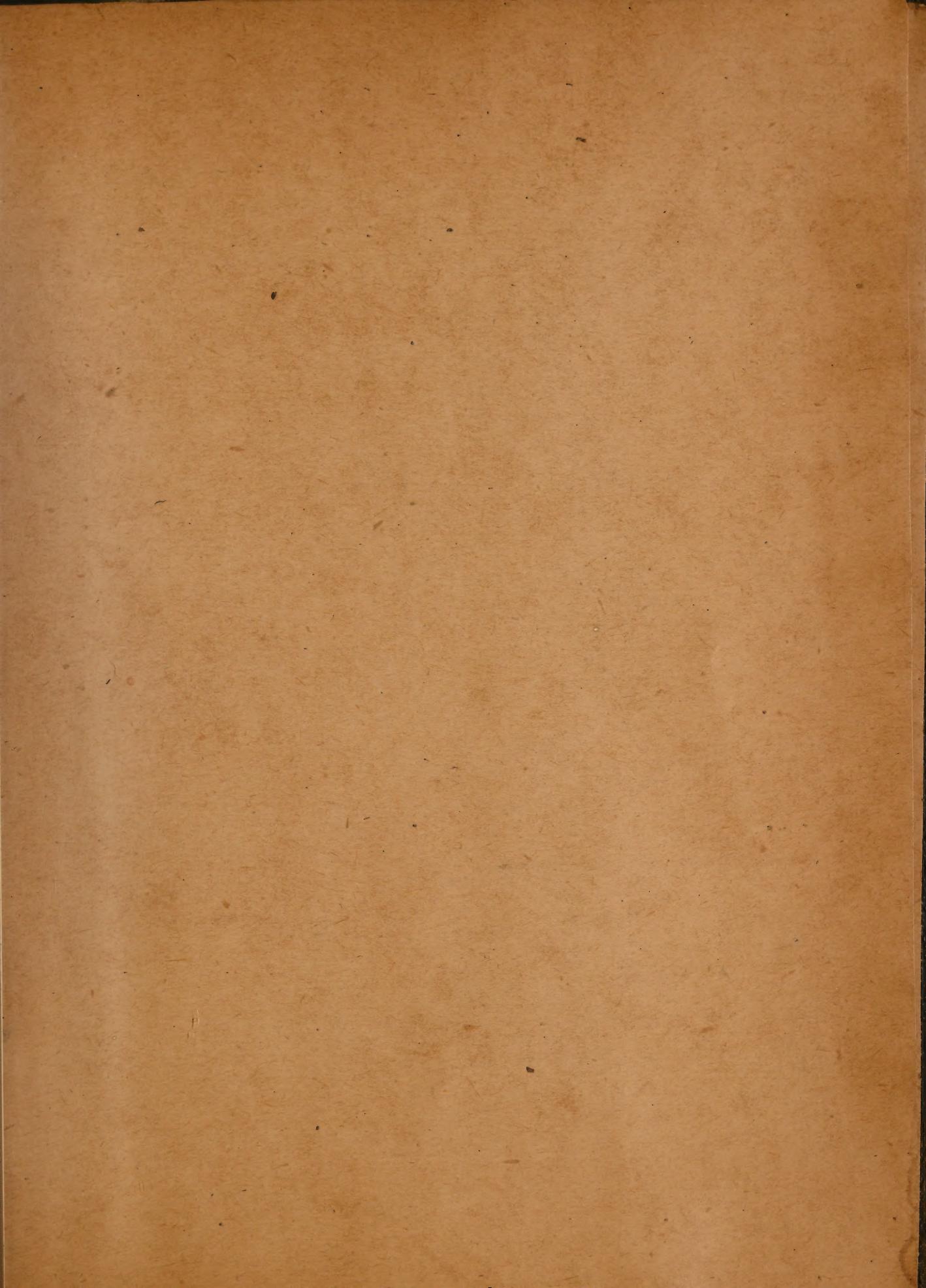
the problem under our
present consideration.











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